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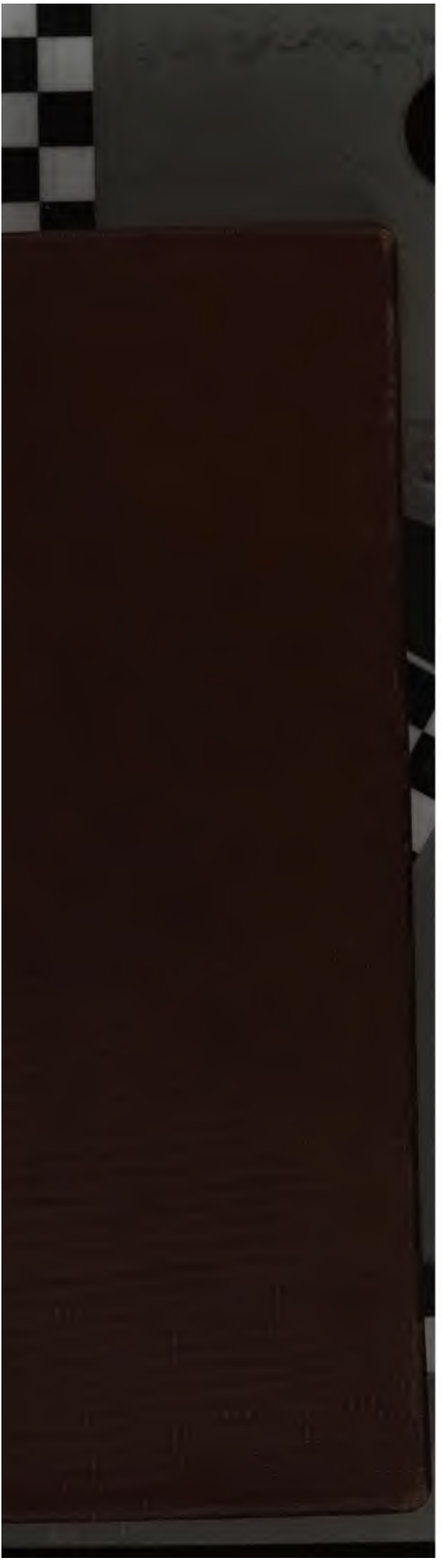
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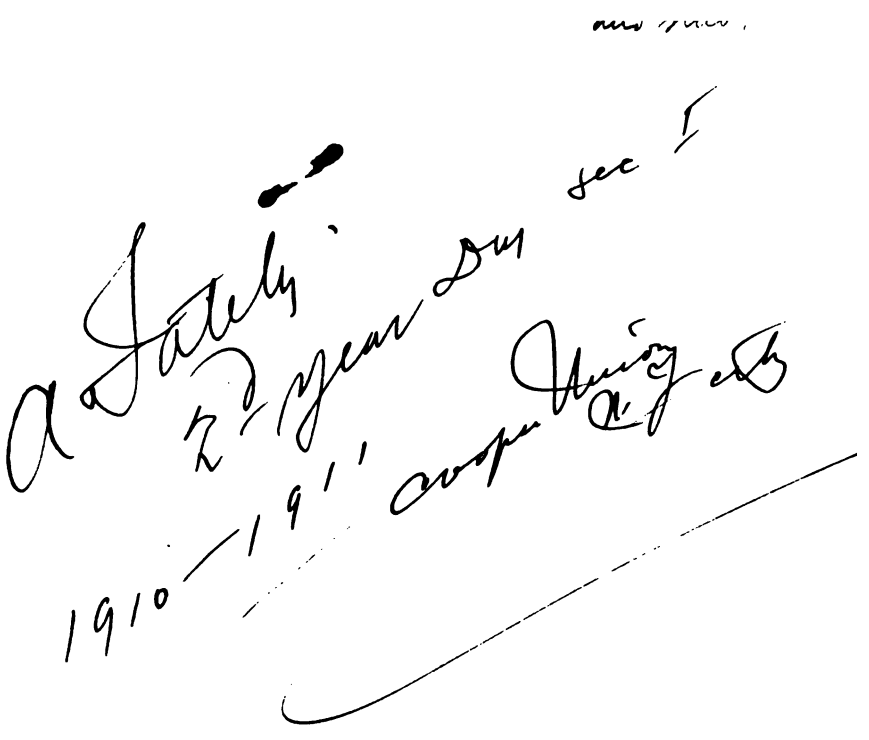
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WORKS OF JOHN S. REID

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Mechanical Drawing and Elementary Machine Design.

By John S. Reid, Professor of Mechanical Drawing, Armour Institute of Technology, and David Reid, formerly Instructor in Mechanical Drawing, Sibley College, Cornell University. Second Edition Revised and Enlarged. 8vo, xii + 439 pages, 329 figures. Cloth, $3.00.

A TEXT-BOOK

OF

MECHANICAL DRAWING

AND

ELEMENTARY

MACHINE DESIGN.

BY

JOHN S. REID,

Instructor in Mechanical Drawing and Machine Design,
Armour Institute of Technology;
Member of the American Society of Mechanical Engineers;

AND

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PREFACE TO THE SECOND EDITION.

THE increasing use of this book for students of all kinds of engineering, in teaching machine drawing and elementary machine design, and the changes which have naturally occurred since the first edition was published, have made a revision and the following additions both necessary and desirable.

Course II has been prepared for students in machine drawing and elementary machine design who have completed the full course given in "A Course in Mechanical Drawing," by John S. Reid, published by John Wiley & Sons, New York, or its equivalent.

The number of problems, their scale and size, have been selected to properly fill a certain size of sheet, together with notes, bill of material, and title. New cuts have been introduced illustrating bills of material and titles, with dimensions for construction.

A decidedly new feature has been introduced here in giving the minimum time allowed for finishing each plate according to directions in the text, and should be much appreciated by Instructors when determining the amount of work to require from their students in any given term. The time allowed to finish the different plates has been carefully determined, and any

student of fair intelligence who will honestly try can finish any of the plates in the time given.

The Course in Lettering has been added for the benefit of students who have not had its equivalent in their preparatory course in mechanical drawing.

Course III is a short course which has been added here as a supplement to Course II, and consists of practical machine sketching, the making of working drawings from sketches, isometrical drawing of a lay-out of piping, and machine design.

The report on the "Present Practice in Drafting Room Methods," which will be found at the end of the book, is also new, and will interest Instructors and enable them to adopt a system in their drawing courses that may closely approximate the best practice in the leading and most progressive drafting rooms in the United States.

The thanks of the authors are due and are most cordially extended to those who have used this book in the past and have encouraged and assisted them by gracious words and timely suggestions.

JOHN S. REID.
D. REID.

ARMOUR INSTITUTE OF TECHNOLOGY,
Chicago, Ill., September, 1908.

PREFACE.

To properly prepare students for advanced machine design it has been found necessary to introduce a course designed to apply the principles of mechanical drawing to the solution of practical problems in machine construction and to familiarize the student with the arrangement and proportions of the most important machines and their details recognized by competent engineers to be the best practice of the present time.

It is essential to intelligent study and an economical expenditure of time and labor that, before attempting to design a new machine or improve an old one, the student should post himself with all possible information concerning what has *already been done* in the same direction.

To this end the present work has been prepared. In it we have attempted to show what is the best United States practice in the design and construction of various machines and details of machines, using rules and formulæ whenever feasible in working out practical problems.

In addition to this will be found the latest and most approved drafting-room methods in use in this country, without which most drawings would be practically useless. Up to the present time no text-book that we know of has been

28 X 37 2

published in the United States that could in the best way fill the need as explained above.

Books of a somewhat similar nature have been published in Great Britain, showing that the same need has been felt there as here. These books, modified to suit American practice, have been used to some extent in this country because they were the best to be had, but are not by any means all that can be desired for our purpose in their present form.

While preparing this course for the sophomore students in Sibley College the authors endeavored to secure samples of the *actual machines* or *parts of machines* as collateral in illustrating the exercises given in the book, with a result that in our drafting-rooms we have many examples of modern machine construction placed convenient to the students' hands, so that they may examine and handle the *actual thing* itself while solving the problems in drawing and designing. This we believe of great importance in the study of machine design and construction, because few are able to describe a machine even with the assistance of a drawing so well as to enable the student to conceive it in his mind as it actually is.

The preparation necessary for the proper understanding and execution of the problems contained in this book is as follows: use of instruments, instrumental drawings applied to drawing geometrical problems in pencil and ink, thorough knowledge of the conventional lines, hatch-lining and colors for sections, mechanical and free-hand lettering, orthographic projection in the third angle, isometrical drawing—in brief all that is contained in "A Course in Mechanical Drawing," by John S. Reid, published by John Wiley & Sons, New York.

In the preparation of the drawings for this work we are

indebted to many of the leading engineering firms of this and other States, who have kindly supplied us with drawings and *samples* of the latest and best practice of the day. Our thanks are especially due to the Dodge Manufacturing Company, the Detroit Screw Works, the Buckeye Engine Co., the United States Metallic Packing Co., the National Tube Works, the Ridgeway Dynamo & Engine Co., the Murray Gun Works, Henry R. Worthington, Robt. Pool & Sons, the Baldwin Locomotive Works, the Schenectady Locomotive Works, the American Pulley Co., the Hyatt Roller Bearing Co., the MacIntosh and Seymour Engine Co., and many others.

Our acknowledgments are also due to many of the best authorities on the different subjects treated, among which may be mentioned Thurston's "Materials of Construction," A. W. Smith's "Machine Design," Klein's "Machine Design," Unwin's "Machine Design," Barr's "Boilers and Furnaces," Peabody and Miller's "Steam Boilers," Low and Bevis's "Drawing and Designing," John H. Barr's "Kinematics," Thurston's "Steam Boilers," Reuleaux's "Constructor," the "Proceedings of the American Railway Master Mechanics' Association," etc., etc.

J. S. R.
D. R.

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from rules and formulæ, will induce the student to think, and tend to develop any natural ability he may have in this direction. It has been the aim of the authors in the arrangement of problems to accomplish this purpose in the highest degree possible.

The following notes on the complete outfit of instruments and materials should be consulted before buying, because it is very essential to the best results that a good outfit be secured.

The outfit for students in mechanical and machine drawing is as follows:

(1) THE DRAWING-BOARD for academy and freshman work is 16″×21″×½″, the same as that used for free-hand drawing. The material should be soft pine and constructed as shown by Fig. 1.

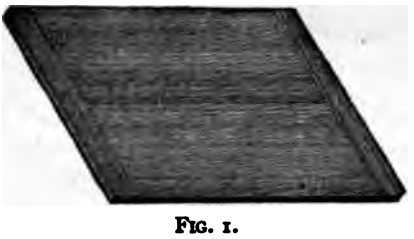


FIG. 1.

(2) 1 SCRIBBLING PENCIL with rubber tip.

(3) PENCILS, one 6H and one 4H Koh-i-noor or Faber.

(4) The T-SQUARE; a plain pearwood T-square with a fixed head is all that is necessary. Length 21″.

(5) INSTRUMENTS. "Pocket Book" Set, shown by Fig. 2, recommended as a first-class medium-priced set of instruments. It contains

A COMPASS, 5½″ long, with fixed needle-point, pencil, pen

and lengthening bar; a SPRING BOW PENCIL, 3″ long; a SPRING BOW PEN, 3″ long; a SPRING BOW SPACER, 3″ long;



FIG. 2.

2 DRAWING-PENS, medium and small, 1 HAIR-SPRING DIVIDER, 5″ long; a nickel-plated box with leads.



FIG. 3.

(6) A TRIANGULAR BOXWOOD SCALE graduated as follows: 4″ and 2″, 3″ and $1\frac{1}{2}$″, 1″ and $\frac{1}{2}$″, $\frac{3}{4}$″ and $\frac{3}{8}$″, $\frac{1}{16}$″ and $\frac{1}{50}$″.

(7) 1 TRIANGLE 30°×60°, celluloid, 10″ long. Fig. 4.

1 " 45°, " 7″ " "

(8) 1 IRREGULAR CURVE. No. 13. Fig. 5.

(9) EMERY PENCIL POINTER.

(10) INK, black waterproof. Fig. 7.

(11) INK ERASER, Faber's Typewriter. No. 104.

(12) PENCIL ERASER, "Emerald" No. 211. Fig. 9.

(13) SPONGE RUBBER or CUBE OF "ARTGUM." Figs. 10, 11.

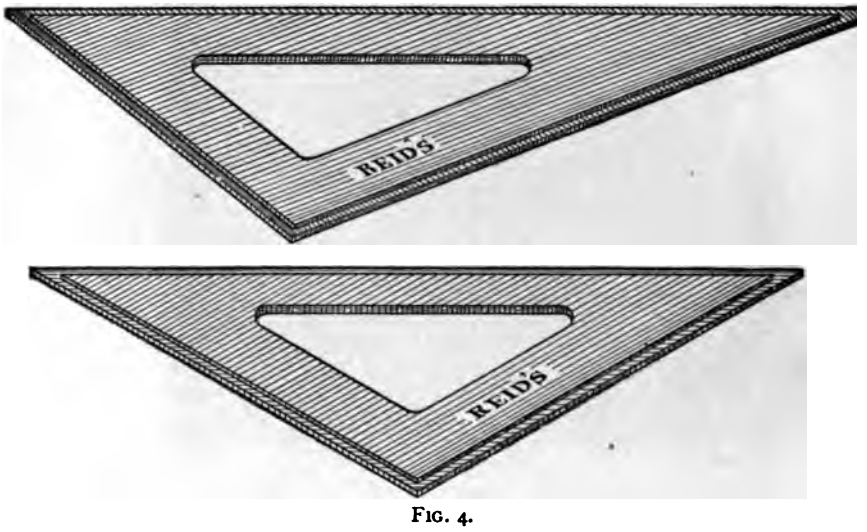


FIG. 4.



FIG. 5.

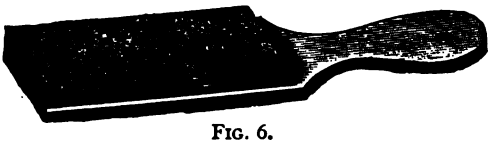


FIG. 6.

(14) TACKS, a small carton of 1 oz. copper tacks, and 1 doz. small thumb tacks.

(15) ARKANSAS OIL STONE. $2'' \times \frac{1}{2}'' \times \frac{1}{16}''$.

(16) PROTRACTOR, German silver, about 5" diam. Fig. 12.

(17) SCALE GUARD, " " Fig. 13.



FIG. 7



FIG. 8.



FIG. 9.



FIG. 10.



FIG. 11.

(18) 2 sheets of "CREAM" DRAWING PAPER. 15"×20".
(19) 2 " " IMPERIAL TRACING CLOTH. 15"×20".
(20) 1 CROSS-SECTION PAD. 8"×10".
(21) 1 SCRIBBLING PAD.

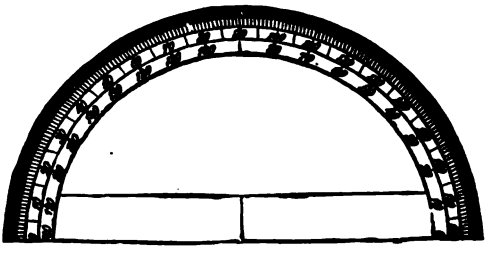


FIG. 12.



FIG. 13.

(22) 1 ERASING SHIELD, nickel plated.
(23) 2 LETTERING PENS, "Gillott" No. 303.
(24) 2 " " "Ball Point," No. 506.
(25) 2 " " " " No. 516.
(26) 1 TWO-FOOT RULE

INSTRUMENTS.

IT is a common belief among students that any kind of cheap instrument will do with which to learn mechanical drawing, and not until they have acquired the proper use of the instruments should they spend money in buying a first-class set. This is one of the greatest mistakes that can be made. Many a student has been discouraged and disgusted because, try as he would, he could not make a good drawing, using a set of instruments with which it would be difficult for even an experienced draftsman to make a creditable showing.

If it is necessary to economize in this direction it is better and easier to get along with a fewer number, and have them of the best, than it is to have an elaborate outfit of questionable quality.

The instruments shown in Fig. 2 are well made of a moderate price, and with care and attention will give good satisfaction for a long time.

USE OF INSTRUMENTS.

The Pencil.—Designs of all kinds are usually worked out in pencil first, and if to be finished and kept they are inked in and sometimes colored and shaded; but if the drawing is only to be finished in pencil, then all the lines except construction, center, and dimension lines should be made broad and dark,

so that the drawing will stand out clear and distinct. It will be noticed that this calls for two kinds of pencil-lines, the first a thin, even line made with a hard, fine-grained lead-pencil, not less than 6H (either Koh-i-noor or Faber's), and sharpened to a knife-edge in the following manner: The lead should be carefully bared of the wood with a knife for about $\frac{1}{2}''$, and the wood neatly tapered back from that point; then lay the lead upon the emery-paper sharpener illustrated in the outfit, and carefully rub to and fro until the pencil assumes a long taper from the wood to the point; now turn it over and do the same with the other side, using toward the last a slightly oscillating motion on both sides until the point has assumed a sharp, thin, knife-edge endwise and an elliptical contour the other way.

This point should then be polished on a piece of scrap drawing-paper until the rough burr left by the emery-paper is removed, leaving a smooth, keen, ideal pencil-point for drawing straight lines.

With such a point but little pressure is required in the hands of the draftsman to draw the most desirable line, one that can be easily erased when necessary and inked in to much better advantage than if the line had been made with a blunt point, because, when the pencil-point is blunt the inclination is to press hard upon it when drawing a line. This forms a groove in the paper which makes it very difficult to draw an even inked line.

The second kind of a pencil-line is the broad line, as explained above; it should be drawn with a somewhat softer pencil, say 4H, and a thicker point.

All lines not necessary to explain the drawing should be

erased before inking or broadening the pencil-lines, so as to make a minimum of erasing and cleaning after the drawing is finished.

When drawing pencil-lines, the pencil should be held in a plane passing through the edge of the T-square perpendicular to the plane of the paper and making an angle with the plane of the paper equal to about 60°.

Lines should always be drawn from left to right. A soft conical-pointed pencil should be used for lettering, figuring, and all free-hand work.

The Drawing-pen.—The best form, in the writer's opinion, is that shown in Fig. 14. The spring on the upper blade



FIG. 14.



FIG. 15.

spreads the blades sufficiently apart to allow for thorough cleaning and sharpening. The hinged blade is therefore unnecessary. The pen should be held in a plane passing through the edge of the T-square at right angles to the plane of the paper, and making an angle with the plane of the paper ranging from 60° to 90°.

The best of drawing-pens will in time wear dull on the point, and until the student has learned from a competent

teacher how to sharpen his pens it would be better to have them sharpened by the manufacturer.

It is difficult to explain the method of sharpening a drawing-pen.

If one blade has worn shorter than the other, the blades should be brought together by means of the thumb-screw, and placing the pen in an upright position draw the point to and fro on the oil-stone in a plane perpendicular to it, raising and lowering the handle of the pen at the same time, to give the proper curve to the point. The Arkansas oil-stones (No. 21 of "The Complete Outfit") are best for this purpose.

The blades should next be opened slightly, and holding the pen in the right hand in a nearly horizontal position, place the lower blade on the stone and move it quickly to and fro, slightly turning the pen with the fingers and elevating the handle a little at the end of each stroke. Having ground the lower blade a little, turn the pen completely over and grind the upper blade in a similar manner for about the same length of time; then clean the blades and examine the extreme points, and if there are still bright spots to be seen continue the grinding until they entirely disappear, and finish the sharpening by polishing on a piece of smooth leather.

The blades should not be too sharp, or they will cut the paper. The grinding should be continued only as long as the bright spots show on the points of the blades.

When inking, the pen should be held in about the same position as described for holding the pencil. Many draftsmen hold the pen vertically. The position may be varied with good results as the pen wears. Lines made with the pen should only be drawn from left to right.

THE TRIANGLES.

The triangles shown at Fig. 4 (in "The Complete Outfit") are 10″ and 7″ long respectively, and are made of transparent celluloid. The black rubber triangles sometimes used are but very little cheaper (about 10 cents) and soon become dirty when in use; the rubber is brittle and more easily broken than the celluloid.

Angles of 15°, 75°, 30°, 45°, 60°, and 90° can readily be drawn with the triangles and T-square. Lines parallel to oblique lines on the drawing can be drawn with the triangles by placing the edge representing the height of one of them so as to coincide with the given line, then place the edge representing the hypotenuse of the other against the corresponding edge of the first, and by sliding the upper on the lower when holding the lower firmly with the left hand any number of lines may be drawn parallel to the given line.

The methods of drawing perpendicular lines and making angles with other lines within the scope of the triangles and T-square are so evident that further explanation is unnecessary.

THE T-SQUARE.

The use of the T-square is very simple, and is accomplished by holding the head firmly with the left hand against the left-hand end of the drawing-board, leaving the right hand free to use the pen or pencil in drawing the required lines.

THE DRAWING-BOARD.

If the left-hand edge of the drawing-board is straight and the T-square, then horizontal lines parallel to the upper edge of the paper and perpendicular to the left-hand edge may be drawn with the T-square, and lines perpendicular to these can be made by means of the triangles, or *set squares*, as they are sometimes called.

THE TRIANGULAR SCALE.

This scale, illustrated in Fig. 3 (in "The Complete Outfit"), was arranged to suit the needs of the students in machine drawing. It is triangular and made of boxwood. The six edges are graduated as follows; $\frac{1}{16}''$ or full size, $\frac{1}{50}''$, $\frac{1}{4}''$ and $\frac{1}{8}'' = 1$ ft., $1''$ and $\frac{1}{2}'' = 1$ ft., $3''$ and $1\frac{1}{2}'' = 1$ ft., and $4''$ and $2'' = 1$ ft.

Drawings of very small objects are generally shown enlarged—e.g., if it is determined to make a drawing twice the full size of an object, then where the object measures one inch the drawing would be made $2''$, etc.

Larger objects or small machine parts are often drawn full size—i.e., the same size as the object really is—and the drawing is said to be made to the scale of full size.

Large machines and large details are usually made to a reduced scale—e.g., if a drawing is to be made to the scale of $2'' = 1$ ft., then $2''$ measured by the standard rule would be divided into 12 equal parts and each part would represent $1''$.

THE SCALE GUARD.

This instrument is shown in No. 17 (in "The Complete Outfit"). It is employed to prevent the scale from turning, so that the draftsman can use it without having to look for the particular edge he needs every time he wants to lay off a measurement.

THE COMPASSES.

When about to draw a circle or an arc of a circle, take hold of the compass at the joint with the thumb and two first fingers, guide the needle-point into the center and set the pencil or pen leg to the required radius, then move the thumb and forefinger up to the small handle provided at the top of

the instrument, and beginning at the lowest point draw the line clockwise. The weight of the compass will be the only down pressure required.

The sharpening of the lead for the compasses is a very important matter, and cannot be emphasized too much. Before commencing a drawing it pays well to take time to properly sharpen the pencil and the lead for compasses and to keep them always in good condition.

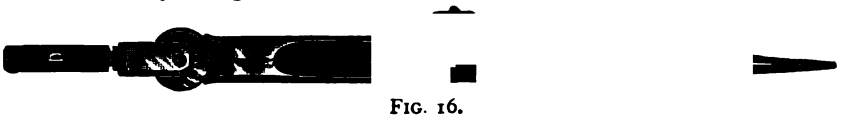


FIG. 16.

The directions for sharpening the compass leads are the same as has already been given for the sharpening of the straight-line pencil.

THE DIVIDERS OR SPACERS.

This instrument should be held in the same manner as described for the compass. It is very useful in laying off equal distances on straight lines or circles. To divide a given line into any number of equal parts with the dividers, say 12, it is best to divide the line into three or four parts first, say 4, and then when one of these parts has been subdivided accurately into three equal parts, it will be a simple matter to step off these latter divisions on the remaining three-fourths of the given line. Care should be taken not to make holes in the paper with the spacers, as it is difficult to ink over them without blotting.

THE SPRING BOWS.

These instruments are valuable for drawing the small circles and arcs of circles. It is very important that all the small arcs, such as fillets, round corners, etc., should be carefully pencilled in before beginning to ink a drawing. Many

good drawings are spoiled because of the bad joints between small arcs and straight lines.

When commencing to ink a drawing, all small arcs and small circles should be inked first, then the larger arcs and circles, and the straight lines last. This is best, because it is

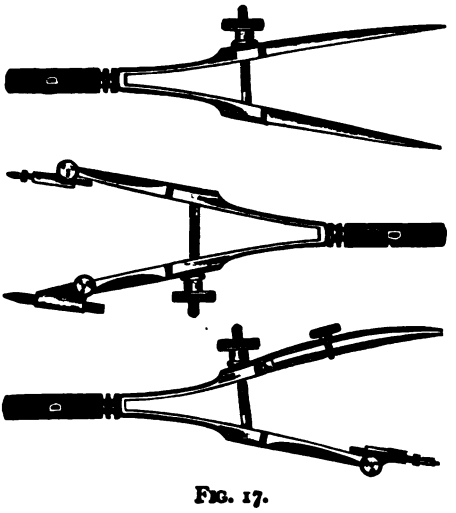


FIG. 17.

much easier to know where to stop the arc line, and to draw the straight line tangent to it, than *vice versa*.

IRREGULAR CURVES.

The irregular curve shown in Fig. 5 is useful for drawing irregular curves through points that have already been found by construction, such as ellipses, cycloids, epicyloids, etc., as in the cases of gear-teeth, cam outlines, rotary pump wheels, etc.

When using these curves, that curve should be selected that will coincide with the greatest number of points on the

THE PROTRACTOR.

This instrument is for measuring and constructing angles. It is shown in Fig. 12. It is used as follows when measuring an angle: Place the lower straight edge on the straight line which forms one of the sides of the angle, with the nick exactly on the point of the angle to be measured. Then the number of degrees contained in the angle may be read from the left, clockwise.

In constructing an angle, place the nick at the point from which it is desired to draw the angle, and on the outer circumference of the protractor, find the figure corresponding to the number of degrees in the required angle, and mark a point on the paper as close as possible to the figure on the protractor; after removing the protractor, draw a line through this point to the nick, which will give the required angle.

SHADE LINES AND SHADING.

Shade Lines are quite generally used on engineering working drawings; they give a relieving appearance to the projecting parts, improve the looks of the drawing and make it easier to read, and are quickly and easily applied.

The *Shading* of the curved surfaces of machine parts is sometimes practiced on specially finished drawings, but on working drawings most employers will not allow shading because it takes too much time, and is not essential to a quick and correct reading of a drawing, especially if a system of shade lines is used.

The Source of Light is considered to be at an infinite distance from the object, therefore the Rays of Light will be represented by parallel lines.

The Source of Light is considered to be fixed, and the Point of Sight situated in front of the object and at an infinite dis-

tance from it, so that the *Visual Rays* are parallel to one another and per. to the plane of projection.

Shade Lines divide illuminated surfaces from dark surfaces.

Dark surfaces are not necessarily to be defined by those surfaces which are darkened by the shadow cast by another part of the object, but by reason of their location in relation to the rays of light.

It is the general practice to shade-line the different projections of an object as if each projection was in the same plane—e.g., suppose a cube, Fig. 18, situated in space in the third angle, the point of sight in front of it, and the direction

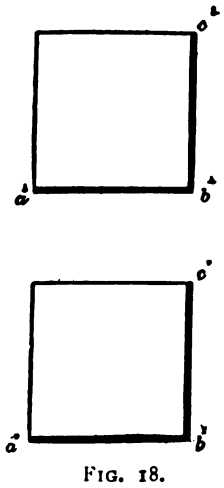


FIG. 18.

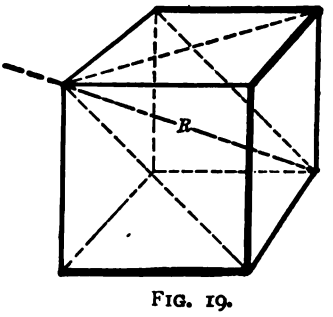


FIG. 19.

of the rays of light coinciding with the diagonal of the cube, as shown by Fig. 19. Then the edges $a^v b^v$, $b^v c^v$ will be shade lines, because they are the edges which separate the illuminated faces (the faces upon which fall the rays of light) from the shaded faces, as shown by Fig. 19.

Now the source of light being fixed, let the point of sight

remain in the same position, and conceive the object to be revolved through the angle of 90° about a hor. axis so that a plan at the top of the object is shown above the elevation, and as the projected rays of light falling in the direction of the diagonal of a cube make angles of 45° with the hor., then with the use of the 45° triangle we can easily determine that the lower and right-hand edges of the plan as well as of the elevation should be shade lines.

This practice then will be followed in this work, viz.:

Shade lines shall be applied to all *projections* of an object, considering the rays of light to fall upon each of them, from the same direction.

Shade lines should have a *width* equal to 3 times that of the other outlines. *Broken lines* should never be shade lines.

The outlines of *surfaces of revolution* should not be shade lines. The shade-lined figures which follow will assist in illustrating the above principles; they should be studied until understood.

WORKING DRAWINGS.

Working drawings are sometimes made on brown detail-paper in pencil, traced on tracing-paper or cloth, and then blue printed.

The latter process is accomplished as follows:

The tracing is placed face down on the glass in the printing-frame, and the prepared paper is placed behind it, with the sensitized surface in contact with the back of the tracing.

In printing from a negative the sensitized surface of the prepared paper is placed in contact with the film side of the negative, and the face is exposed to the light.

The blue-print system for working drawings has many drawbacks, e.g., the sectional parts of the drawing requires to be hatch-lined, using the standard conventions already referred to for the different materials. This takes a great deal of time. The print has usually to be mounted on cardboard, although this is not always done, and unless it is varnished the frequent handling with dirty, oily fingers soon makes it unfit for use.

Changes can be made on the prints with soda-water, it is true, but they seldom look well, and when many changes or additions require to be made it is best to make them on the tracing and take a new print. And the sunlight is not always favorable to quick printing. So taking everything into consideration the system of making working drawings directly on cards and varnishing them is probably the best. It is the system used by the Schenectady Locomotive Works and many other large engineering establishments. In size the cards are made $9'' \times 12''$, $12'' \times 18''$, $18'' \times 24''$; they are made of thick pasteboard mounted with Irish linen record-paper. The drawings are pencilled and inked on these cards in the usual way, and the sections are tinted with the conventional colors, which are much quicker applied than hatch-lines. The face of the drawing is protected with two coats of white shellac varnish, while the back of the card is usually given a coat of orange shellac.

The white varnish can easily be removed with a little alcohol, and changes made on the drawing, and when revarnished it is again ready for the shop.

In the hands of an experienced workman a working drawing is intended to convey to him all the necessary

information as to shape, size, material, and finish to enable him to properly construct it without any additional instructions. This means that it must have a sufficient number of elevations, sections, and plans to thoroughly explain and describe the object in every particular. And these views should be completely and conveniently dimensioned. The dimensions on the drawing must of course give the sizes to which the object is to be made, without reference to the scale to which it may be drawn. The title of a working drawing should be as brief as possible, and not very large—a neat, plain, free-hand printed letter is best for this purpose.

Finished parts are usually indicated by the letter "f," and if it is all to be finished, then below the title it is customary to write or print "finished all over."

The number of the drawing may be placed at the upper left-hand corner, and the *initials* of the draftsman immediately below it.

Lettering.—All lettering on mechanical drawings should be plain and legible, but the letters in a title or the figures on a drawing should never be so large as to make them appear more prominent than the drawing itself.

The best form of letter for practical use is that which gives the neatest appearance with a maximum of legibility and requires the least amount of time and labor in its construction.

Figuring.—Great care should be taken in figuring or dimensioning a mechanical drawing, and especially a working drawing.

To have a drawing accurately, legibly, and neatly figured is considered by practical men to be the most important part of a working drawing.

There should be absolutely no doubt whatever about the character of a number representing a dimension on a drawing.

Many mistakes have been made, incurring loss in time, labor, and money through a wrong reading of a dimension.

Drawings should be so fully dimensioned that there will be no need for the pattern-maker or machinist to measure any part of them. Indeed, means are taken to prevent him from doing so, because of the liability of the workman to make mistakes, so drawings are often made to scales which are difficult to measure with a common rule, such as 2″ and 4″ = 1 ft.

STANDARD CONVENTIONAL SECTION LINES.

Conventional section lines are placed on drawings to distinguish the different kinds of materials used when such drawings are to be finished in pencil, or traced for blue printing, or to be used for a reproduction of any kind.

Water-colors are nearly always used for finished drawings and sometimes for tracings and pencil drawings.

The color tints can be applied in much less time than it takes to hatch-line a drawing. So that the color method should be used whenever possible.

To apply the color tint.—Great care should be taken in determining the depth of the tint to be used; when only the section parts are to be colored the tints should be quite *light* because it is much easier to obtain an even wash and a softer and more artistic effect. Before applying the color the drawing board should be cleared of drawing instruments, etc., so that it may be easily turned to enable the student to keep

the bounding color line always to his left, and keeping the brush in such position that the color just touches the bounding line transfer the color to the drawing with long sweeps of the brush until the surface is covered. Press out all color remaining in the brush with the fingers and apply the brush again to the little puddles remaining on the paper. The brush will draw it back into itself and leave an even tint all over the section.

FIG. 20.—This figure shows a collection of hatch-lined sections that is now the almost universal practice among draftsmen in this and other countries, and may be considered standard.

No. 1. To the right is shown a section of a wall made of *rocks*. When used without color, as in tracing for printing, the rocks are simply shaded with India ink and a 175 Gillott steel pen. For a colored drawing the ground work is made of gamboge or burnt umber. To the left is the conventional representation of water for tracings. For colored drawings a blended wash of Prussian blue is added.

No. 2. *Convention for Marble.*— When colored, the whole section is made thoroughly wet and each stone is then streaked with Payne's gray.

No. 3. *Convention for Chestnut.*— When colored, a ground wash of gamboge with a little crimson lake and burnt umber is used. The colors for graining should be mixed in a separate dish, burnt umber with a little Payne's gray and crimson lake added in equal quantities and made dark enough to form a sufficient contrast to the ground color.

No. 4. *General Convention for Wood.*—When colored the ground work should be made with a light wash of burnt sienna.

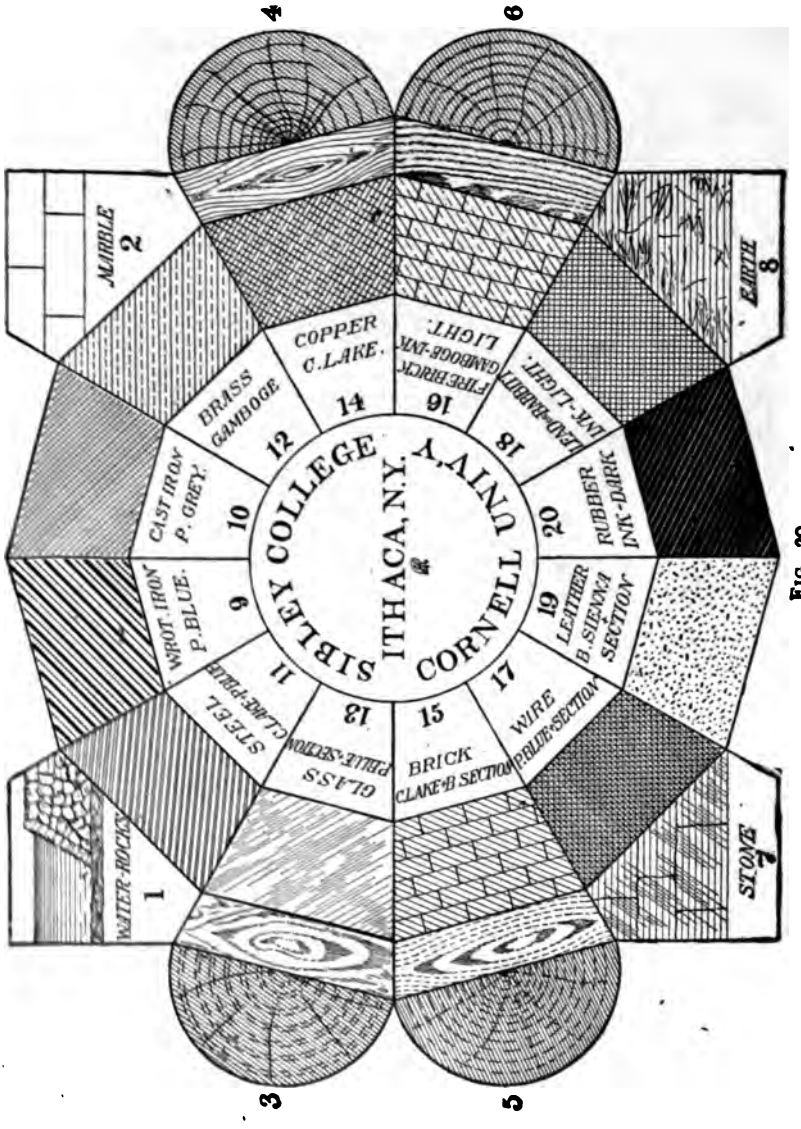

SIBLEY COLLEGE
CORNELL UNIV'Y
ITHACA, N.Y.
1 WATER-ROCKS
2 MARBLE
3
4
5
6 EARTH
7 STONE
8
9 WROT. IRON P. BLUE.
10 CAST IRON P. GREY
11 STEEL CLARET-PRUSS
12 BRASS GAMBOGE
13 GLASS PALE-SEPIA
14 COPPER C. LAKE.
15 BRICK CLAKE-B SECTION
16 FIRE BRICK GAMBOGE-INK LIGHT
17 WIRE PRUSS-SECTION
18 LEAD-RABBIT INK-LIGHT
19 LEATHER B. SIENNA SECTION
20 RUBBER INK-DARK

FIG. 20.

The graining should be done with a writing-pen and a dark mixture of burnt sienna and a modicum of India ink.

No. 5. *Convention for Black Walnut.*—A mixture of Payne's gray, burnt umber and crimson lake in equal quantities is used for the ground color. The same mixture is used for graining when made dark by adding more burnt umber.

No. 6. *Convention for Hard Pine.*—For the ground color make a light wash of crimson lake, burnt umber, and gamboge, equal parts. For graining use a darker mixture of of crimson lake and burnt umber.

No. 7. *Convention for Building-stone.*—The ground color is a light wash of Payne's gray and the shade lines are added mechanically with the drawing-pen or free-hand with the writing-pen.

No. 8. *Convention for Earth.*—Ground color, India ink and neutral tint. The irregular lines to be added with a writing-pen and India ink.

No. 9. *Section Lining for Wrought or Malleable Iron.*—When the drawing is to be tinted, the color used is Prussian blue.

No. 10. *Cast Iron.*—These section lines should be drawn equidistant, not very far apart and narrower than the body lines of the drawing. The tint is Payne's gray.

No. 11. *Steel.*—This section is used for all kinds of steel. The lines should be of the same width as those used for cast-iron and the spaces between the double and single lines should be uniform. The color tint is Prussian blue with enough crimson lake added to make a warm purple.

No. 12. *Brass.*—This section is generally used for all kinds of composition brass, such as gun-metal, yellow metal,

bronze metal, Muntz metal, etc. The width of the full lines, dash lines and spaces should all be uniform. The color tint is a light wash of gamboge.

Nos. 13–20.—The section lines and color tints for these numbers are so plainly given in the figure that further instruction would seem to be superfluous.

Sometimes draftsmen will crosshatch all the sectional parts with a uniform space and line like that used for cast iron and mark the names of the different materials or their initials in some convenient place on the parts themselves. This does not look as well nor is it any more convenient to experienced men than the other method.

CONVENTIONAL LINES.

FIG. 21.—There are four kinds:

(1) *The Hidden Line.*—This line should be made of short dashes of uniform length and width, both depending somewhat on the size of the drawing. The width should always be slightly less than the body lines of the drawing, and the

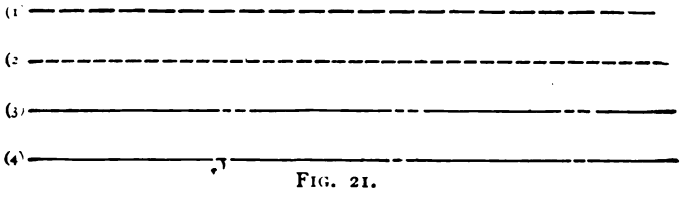


FIG. 21.

length of the dash should never exceed $\frac{1}{8}''$. The spaces between the dashes should all be uniform, quite small, never exceeding $\frac{1}{16}''$. This line is always inked in with *black ink.*

(2) *The Line of Motion.*—This line is used to indicate *point paths.* The dashes should be made shorter than those of the *hidden line*, just a trifle longer than dots. The spaces should of course be short and uniform.

(3) *Center Lines.*—Most drawings of machines and parts of machines are symmetrical about their center lines. When penciling a drawing these lines may be drawn continuous and as fine as possible, but on drawings for reproductions the black-inked line should be a long narrow dash and two short ones alternately. When colored inks are used the center line should be made a continuous *red* line and as fine as it is possible to make it.

(4) *Dimension Lines and Line of Section.*—These lines are made in *black* with a fine long dash and one short dash alternately. In color they should be continuous *blue* lines. Colored lines should be used wherever feasible, because they are so quickly drawn and when made fine they give the drawing a much neater appearance than when the conventional black lines are used. Colored lines should *never* be broken.

CONVENTIONAL BREAKS.

FIG. 22.—Breaks are used in drawings sometimes to indicate that the thing is actually longer than it is drawn, some-

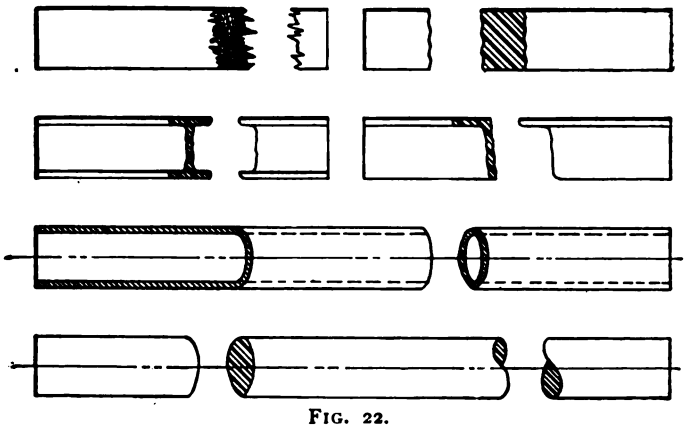


FIG. 22.

times to show the shape of the cross-section and the kind of material. Those given in Fig. 22 show the usual practice.

CROSS-SECTIONS.

FIG. 23.—When a cross-section of a pulley, gear-wheel or other similar object is required and the cutting-plane passes through one of the spokes or arms, then only the rim and hub should be sectioned, as shown at *xx* No. 1 and *zz* No. 2, and the arm or spoke simply outlined. Cross-sections of the arms may be made as shown at *AA* No. 2. In working drawings of gear-wheels only the number of teeth included in one quadrant need be drawn; the balance is usually shown by conventional lines, e.g., the *pitch* line the same as a center line, viz., a long

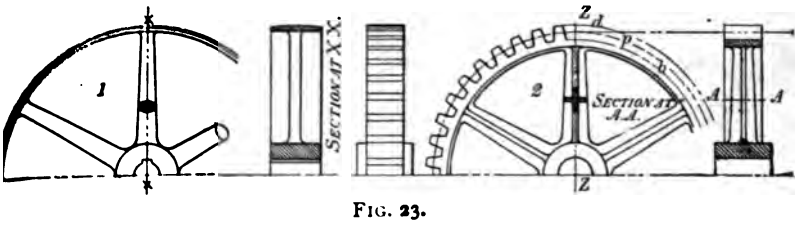


FIG. 23.

dash and two very short ones alternately or a fine continuous *red* line.

The *addendum* line (*d*) and the *root* or *bottom line* (*b*) the same as a dimension line, viz., one long dash and one short dash alternately or a fine continuous *blue* line. The end elevation of the gear-teeth should be made by projecting only the points of the teeth, as shown at No. 2.

Other conventions will be referred to in the text connected with the figures in which they are illustrated.

Constructions.—To draw the curve of intersection that is formed by a plane cutting an irregular surface of revolution.

Figs. 24 and 25 show examples of engine connecting-rod ends where the curve *I* is formed by the intersection of

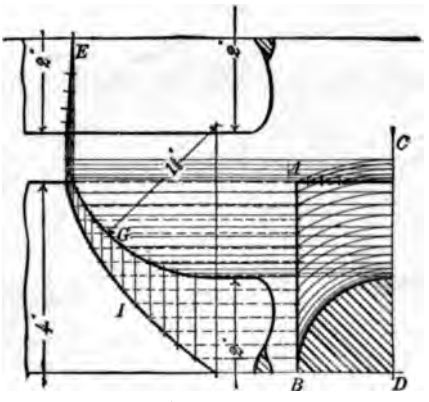


FIG. 24.

the flat stub end with the surface of revolution of the turned part of the rod.

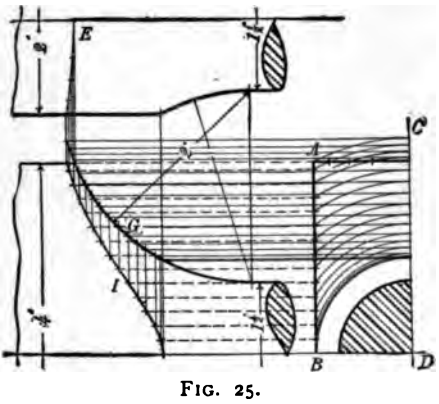


FIG. 25.

Divide the line *AB*, Figs. 24 and 25, into any number of equal parts and through them describe arcs cutting the center line *CD*. Through the intersections of these arcs with *CD* draw horizontals to intersect the curve or fillet *G*.

Through the intersections on G draw perpendiculars and from the divisions on AB draw horizontals to intersect the perpendiculars; these latter intersections are points in the curve I.

The curve E can be found in a similar way as shown by the figure.

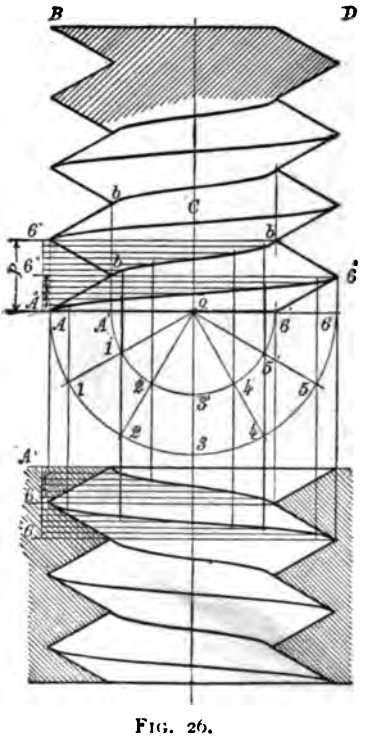


FIG. 26.

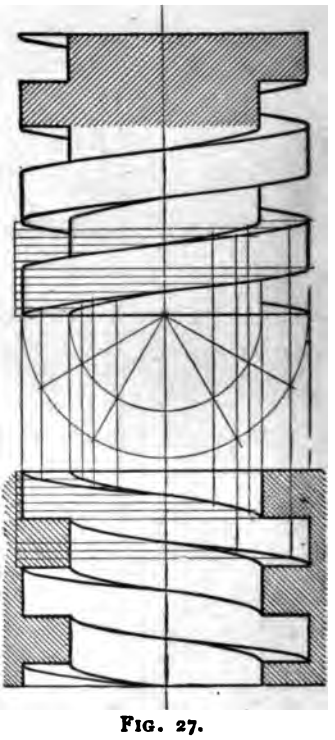


FIG. 27.

To draw the projections of a V-threaded screw and its nut of 3″ diam. and $\frac{3}{4}$″ pitch.

Begin by drawing the center line C, Fig. 26, and lay off on each side of it the radius of the screw $1\frac{1}{2}$″. Draw $A6$ and $6D$. Draw $A6$ the bottom of the screw, and on AB step off the pitch $= \frac{3}{4}$″, beginning at the point A.

On line $6D$ from the point 6 lay off a distance = half the pitch = $\frac{3}{8}''$, because when the point of the thread has completed half a revolution it will have risen perpendicularly a distance = half the pitch, viz., $\frac{3}{8}''$.

Then from the point $6''$ on $6D$ step off as many pitches as may be desired. From the points of the threads just found, draw with the 30° triangle and T-square the V of the threads intersecting at the points $b\,.\,.\,b\,.\,.$ the bottom of the threads.

At the point O on line $A6$ draw two semicircles with radii = the top and bottom of the thread respectively. Divide these into any number of equal parts and also the pitch P into the same number of equal parts. Through these divisions draw hors. and pers. intersecting each other in the points as shown by Fig. 26, which shows an elevation partly in section and a section of a nut to fit the screw. Through the points of intersection draw the curves of the helices shown, using No. 3 of the "Sibley College Set" of Irregular Curves.

ELEMENTARY MACHINE DESIGN.

A machine, according to Prof. John H. Barr, is "a combination of resistant bodies for modifying energy and doing work, the members of which are so arranged that, in operation, the motion of any member involves definite, relative, constrained motion of the others."

In order to obtain the most desirable results in designing such a structure it is necessary to give the several bodies composing it such form and proportion as will enable them to perform their functions in the best possible way and at the same time present a pleasing appearance to the experienced

eye. And, moreover, it must not be forgotten that these desired results should be sought with a due regard to *economy* of material and construction.

The *form* of a machine will probably depend largely upon the designer's experience and his natural ability or intuition.

The proportion of the several parts may be calculated if the opposing forces are known, but in many cases these forces cannot be accurately determined and the designer must rely upon the most approved practice of the past had under similar conditions.

MATERIALS USED IN MACHINE CONSTRUCTION.

The principal materials used in machine construction may be divided into three heads, viz.: Cast Metals, Wrought Metals, and Wood.

CAST METALS.

Among the cast metals the more important in machine construction are cast iron, malleable cast iron, cast steel, brass, copper-bronze or gun-metal, phosphor-bronze, and aluminum.

Cast Iron.—Three kinds of white cast iron and three of gray are used in different ways in machine construction. The whitest iron is very hard and is used like the others of its class for making wrought iron.

The gray irons do not melt as readily as the white, but are more fluid when melted. The grayest irons are the weakest and are used only for mixing with others in the cupola.

Ordinary cast iron contains from 3% to 5% of carbon, which in the white iron is fully combined with the iron, while only .6% to 1.5% is combined in the gray iron and 2.9% to 3.7% shows as graphite crystals.

Iron castings of machine-parts are made from patterns. These patterns are made of wood, usually soft pine, in form exactly like the castings desired. The patterns are used to make moulds in sand in the foundry and into these moulds is poured the molten iron.

Cast iron after solidifying in the moulds contracts while cooling about $\frac{1}{8}''$ per foot of length. To allow for this contraction pattern-makers use a special rule called a shrink-rule for measuring patterns; it is $\frac{1}{8}''$ per foot longer than the standard rule.

Sharp corners in patterns do not cast sharp and square in the metal, but come out ragged and blunt, so that whenever possible *sharp edges should be rounded* and *sharp concave corners filleted or partially filled in;* the result is a stronger and better-looking casting.

To avoid irregular internal strains in iron castings when cooling it is necessary that the section of the casting be made as uniform as possible, so that the metal may contract uniformly throughout.

Chilled Castings.—Melted gray cast iron if cooled quickly retains in chemical combination a large amount of carbon which otherwise would be separated from the casting. The result is a white hard iron called chilled cast iron. To secure this quick cooling the mould into which the metal is cast is made of thick cast iron, which draws the heat from the molten metal in much less time than does the sand mould.

Malleable Castings are made by putting a gray-iron casting in a suitable box and covering it with powdered red hematite, which is an oxide of iron, and keeping it in a furnace at a bright-red heat for from two to thirty hours or even longer, depending upon the size of the casting; such castings are valuable for small light parts of machines, because they are tough and strong. Malleable castings can be worked like wrought iron, but will not weld.

Cast Steel is made by melting broken pieces of blister-steel in a closed crucible and casting into ingots.

Brass is very much used, because it is easy to work, is cheap, strong, and tough, and of a good color. The usual composition of brass is 2 of copper to 1 of zinc, with sometimes a little lead added.

Muntz Metal is a brass composition of 3 parts copper to 2 of zinc. It can be rolled or forged when hot and is used in the shape of bolts and nuts, sheets for sheathing wooden vessels, and often takes the place of iron or steel because of its ability to withstand the corrosive action of water.

Copper.—Pure copper with a small addition of phosphorus makes fairly good castings, but it is difficult to obtain sound castings from copper alone. Copper has a reddish-brown color and is very malleable and ductile when pure. It can be hammered, rolled, and forged when hot or cold; joints can be united by brazing, but welding is difficult. The annealing of iron and steel is effected by heating and slow cooling, while copper can only be annealed by heating and quick cooling.

Bronze or Gun-metal.—The best composition is made of 9 parts of copper to 1 of tin. For bearings designed to sus-

tain great pressure very hard bronze is often used, in which the proportion of tin is increased to 14 parts with 86 parts of copper.

Phosphor-bronze.—This alloy is made by adding from 2% to 4% of phosphorus to the common bronze. It is used for many things in place of iron and steel, such as pump-rods, ship-propellers, etc.; it is also used quite largely for locomotive axle-bearings and shows excellent wearing qualities.

Babbitt Metal.—This is a soft white metal that is used quite largely for lining shaft-bearings. Its composition is usually as follows: copper 4 parts, antimony 8, tin 24, melted together, and before using this alloy is melted with an addition of twice its weight of tin and applied to the bearings while molten. So the real composition of the lining is copper 4, antimony 8, and tin 96.

Aluminum.—This is a very light metal, soft, malleable, and ductile, and of a silvery-white color with a bluish tint. A process for producing it with comparative cheapness was discovered in 1890, and since then its production has been rapidly increasing. It is thoroughly non-corrosive.

WROUGHT METALS.

These consist of wrought iron and steel of various qualities.

Wrought Iron or Malleable Iron is a white metal not easily melted and is very strong and tough. It is made from the white cast irons by abstracting the most of the latter's carbon in a puddling-furnace. It is taken from this furnace in large spongy masses called blooms, and shingled by repeated squeezing and hammering and rolled into what is known as puddled bars. The puddled bars are then cut into short

pieces and piled into faggots; these are heated again and rolled into what is known as merchant bars. The best qualities of wrought iron are piled together, reheated, and rolled in the same way many times, giving the iron its fibrous nature which makes it so tough and strong. A valuable property of wrought iron is that it can be welded at a temperature of from 1500° to 1600° Fahr.

Case-hardening.—This is a hardening of the surface of finished parts of machines, such as the links, guides, etc., of steam-engines, so that their wearing qualities are very much increased. It is effected as follows: the piece to be case-hardened is placed in a suitable receptacle and surrounded by bone-dust, horn-shavings, yellow prussiate of potash, or any such substance that is rich in carbon, and heated to about a red heat, when the wrought iron will absorb some of the carbon surrounding it and be converted into steel, which can be hardened by immersing in water.

Steel is made from wrought iron by adding a little carbon or from cast iron by extracting some of its carbon. There are three ways of doing this: the Bessemer, Siemens-Martin, and cementation processes.

Bessemer Steel is made by pouring melted cast iron into a converter through which a blast of air is forced. In this way the carbon in the cast iron is burnt out, leaving almost pure iron. To this is added a certain quantity of spiegeleisen, which is a compound of iron, carbon, and manganese, and then the molten metal is cast into steel ingots.

Siemens-Martin Steel is made by melting wrought iron and cast iron, or cast iron and certain kinds of iron ore, together on the hearth of a reverberatory gas-furnace.

The *Cementation Process* consists of embedding bars of wrought iron in powdered charcoal in a fire-clay trough and placed in a furnace for several days at a high temperature. The iron combines with portions of the carbon and forms blister-steel, so called from the blisters found on its surface. Bars of blister-steel about 18″ long are then bound together by strong steel wire and heated to a welding heat, then hammered and rolled into bars called shear-steel.

WOODS.

The woods used in machine construction are principally pine, fir, beech, boxwood, ash, elm, hornbeam, lignum-vitæ, mahogany, oak, and teak.

Pine and Fir are strong, cheap, and easy to work, and are largely used for a variety of purposes.

White and *Yellow Pine* are much used in pattern-making.

Beech is used for the cogs of mortise-wheels; it takes a smooth surface and is very close-grained.

Boxwood is much used for sheaves of pulley-blocks and bearings. It takes a smooth surface, is hard, heavy, and of a bright-yellow color.

Elm is very durable in water, and is therefore used for paddle-wheel floats, piles, etc.

Hornbeam is often used for cogs of mortise-wheels.

Lignum-vitæ. — This is a very hard wood of great strength and durability under water. For these reasons it is used for bearings under water and other purposes requiring hardness and strength. Its specific gravity is 1.33; i.e., $1\frac{1}{3}$ times the weight of the same volume of water.

Mahogany is a favorite for making small patterns. It is

straight-grained, strong, and durable, and does not as readily change its form when seasoning as most other woods.

Oak is tough and straight-grained, very durable, whether used dry or in water. It is used for machine-framing and supports.

Teak is a strong, tough, durable wood. It shrinks very little when seasoning, and is very valuable on that account. Bolts passing through it are prevented from rusting by the oil it contains.

STRENGTH OF MATERIALS.

DEFINITIONS.

Load.—The load on any member of a machine is a total of the external forces acting on it. The useful load is the load which the member is designed to carry outside of itself; e.g., the useful load on the springs of a railway-car is the load which may be placed upon the car in addition to the load arising from the weight of the car itself. A live load is a variable load applied and removed continuously. A dead load or constant load is that which has an unvarying and continuous straining action.

Strain and Stress.—Strain is the change of form produced by the action of a load. If the load does not exceed the elastic limit of the material the strain will disappear when the load is removed. Machine-members should be designed strong enough to resist permanent set under maximum load.

Stress is the force which causes strain. The different kinds of stress are: *tensile* stress or pull, *compressive* stress or

thrust, *shearing* stress or cross-cutting, *bending* or combined thrust and pull, and *torsional* or twisting stress.

Resistance of metal to change of form is due to the inherent cohesive force of its molecules.

Elasticity or spring is the inherent property in a material of regaining original form after an external load has been removed.

Elastic Limit.—The elastic limit is the limit of extension or compression to which a material can be subjected without permanent set. Within the elastic limit strain and stress are proportional.

Modulus of Elasticity. — Dr. Thomas Young of the British Royal Society propounded the following formula for the modulus of elasticity (E) in 1826, known as "Young's Modulus":

$$E = \frac{\text{stress per sq. in. in lbs.}}{\text{strain per inch of length}} \text{ (within the elastic limit).}$$

TABLE 1.

ELASTIC MODULI.

Material.	Modulus of Elasticity.	Material.	Modulus of Elasticity.
Wrought iron, bars....	29,000,000	Pine, average.....	1,500,000
Wrought iron, plates..	26,000,000	Beech.....	1,350,000
Cast steel.....	30,000,000	Boxwood.....	1,800,000
Cast steel, tempered...	36,000,000	Ash.....	1,600,000
Forged steel.....	30,000,000	Elm.....	900,000
Steel plates.....	31,000,000	Lignum-vitæ.....	1,000,000
Cast iron, average.....	17,000,000	Mahogany.....	1,300,000
Brass and bronze.....	12,000,000	Oak, English.....	1,700,000
Muntz metal.....	14,000,000	Teak.....	2,300,000
Copper, average.....	12,000,000	Leather.....	24,500
Lead, sheet.....	720,000	Glass, plate.....	8,000,000

Ultimate Strength is the smallest load that will fracture a member under stress.

The Proof Strength is nearly equal to the load that will cause permanent set; i.e., to the maximum elastic resistance.

The Factor of Safety is the ratio of the ultimate strength of a member to the working load, or the breaking load to the actual load. The factor of safety changes for different materials and for different uses of the same material. It is of course much greater under live loads than under constant dead loads.

The following table gives the ordinary factors of safety in general use:

TABLE 2.

FACTORS OF SAFETY.

Material.	Ratio of Ultimate Load to Working Load.		
	Dead Load.	Live Load.	Shocks.
Cast iron	4	7	15
Wrought iron	3	5 to 8	9 to 13
Mild steel	3	5 to 8	9 to 13
Cast steel	3	5 to 8	10 to 15
Copper and similar metals and alloys	5	8	10 to 15
Wood	8	10	14 to 18
Brick and stone	10	20	30

Strength of Cast Iron.—The average American cast iron has a tenacity of about 20,000 lbs. per sq. in., but cast iron has been made which showed an ultimate tensile strength of 35,000 lbs. per sq. in.

The ultimate *compressive* strength of cast iron is from 4 to 6 times its tenacity,—the average is about 90,000 lbs. per sq. in.,—and the average *shearing* strength is about 20,000 lbs. per sq. in. The elastic limit of cast iron is from $\frac{1}{3}$ to nearly equal to the breaking strength.

Strength of Wrought Iron.—The elastic strength of wrought iron is usually over half its ultimate strength; good bars and plates will show an elastic limit of about 26,000 lbs. In ascertaining the strength of a particular piece of wrought iron it will be necessary to know the elongation per cent of specimen. The elongation is greater for short than for long specimens. The usual length of specimens for tensile test is 8″. Wrought iron loses its strength in forging; this loss of strength is equal to about 20%. The difference between the strength of wrought iron when pulled against the grain and in the direction of the grain is from 3000 to 9000 lbs. per sq. in., the strength in the direction of the grain being the greater. The tensile strength of wrought iron varies from 40,000 to 60,000 lbs. per sq. in.

Strength of Steel.—The steel cast from blister-steel is the strongest, having a tensile strength of from 100,000 to 130,000 lbs. per sq. in., but it is hard and brittle, with an elongation of only about 5%. It is therefore unsuitable for constructive purposes. A good plate steel for steam-boilers has a tensile strength of from 55,000 to 60,000 lbs., with an elongation of about 20% in a length of 8″.

The following tables were compiled after consulting various authorities; e.g., Thurston, Unwin, Kent, Molesworth, etc.

TABLE 3.

AVERAGE ULTIMATE AND ELASTIC STRENGTH OF VARIOUS MATERIALS AN MODULI OF ELASTICITY IN POUNDS PER SQUARE INCH.

Material.	Ultimate Strength.			Elastic Strength.			*E.* With the Grain.	*E'.* Trans verse.
	Tension.	Compression.	Shearing.	Tension.	Compression.	Shearing.		
Cast iron, common.	20,000	90,000	20,000	12,000	23,000	9,000	15,000,000	7,000,0
Wrought iron, bars..	50,000	48,000	40,000	26,000	26,000	22,000	29,000,000	10,500,0
Wrought iron, plates.	48,000	46,000	38,000	26,000	26,000	22,000	29,000,000	10,000,0
Wrought shape iron..	48,000	46,000	38,000	26,000	26,000	22,000	29,000,000	10,000,0
Wrought stay-bolt iron	51,000	49,000	40,000	28,000	28,000	24,000	29,000,000	10,000,0
Wrought rivets	50,000	48,000	38,000	26,000	26,000	22,000	29,000,000	10,000,0
Malleable cast iron..	32,000		38,000	23,500	27,000	20,000	24,500,000	13,500,0
Cast steel	88,000	125,000	64,000	70,000		64,000	30,000,000	11,000,0
Soft-steel plates.....	55,000	66,000	50,000	32,000		25,000	30,000,000	11,000,0
Steel rivets	52,665	50,000	45,000	29,000			29,000,000	10,500,0
Cast copper.....	23,000	45,000		5,900				
Forged copper	34,000	58,000		4,500	4,000	3,000	15,000,000	6,000,0
Brass, yellow.....	17,500	10,500		7,000		5,200	9,200,000	3,500,0
Gun-metal	36,000			6,200		4,200	10,000,000	4,000,0
Wood, pine	12,000	6,000	650				1,600,000	90,0
Wood, oak, English	15,000	10,000	2,300				1,700,000	82,0
Leather.....	4,200						25,000	
Stone		6,000						

Wrought Iron has a specific gravity of 7.5 to 7.8 accor ing to its chemical composition and physical structure.

Cast Iron has a specific gravity of 7.25.

The tensile strength of metals varies with their temper ture, generally decreasing as their temperature is increased.

TABLE 4.

RELATIVE TENACITIES OF METAL. (THURSTON.)

Lead	1.0	Cast iron	7 to 12
Tin	1.3	Wrought iron	20 to 40
Zinc	2.0	Steel	40 to 100
Worked copper	12 to 20		

USEFUL TABLES AND MISCELLANEOUS INFORMATION.

WEIGHTS AND MEASURES.

Avoirdupois or Commercial Weight.

16 drachms	1 ounce.
16 ounces	1 pound.
14 pounds	1 stone.
28 "	1 quarter.
4 quarters	1 cwt.
2240 pounds	1 ton.

Square Measure.

144	square inches	1	square foot.
9 "	feet	1 "	yard.
30¼ "	yards	1 "	rod.
40 "	rods	1 "	rood.
4 "	roods	1 "	acre.
640 "	acres	1 "	mile.

Measure of Volume.

A cubic foot has	1728 cubic inches.
An ale gallon has	282 " "
A standard or wine gallon has	231 " "
A dry gallon has	268.8 " "
A bushel has	2150.4 " "
A cord of wood has	128 " feet.
A perch of stone has	24.75 " "
A ton of round timber has	40 " "
A " hewn " "	50 " "
A box 19⅝ × 19⅝ inches, 19⅝ inches deep, contains	1 barrel.
A " 12$\frac{45}{56}$ × 12$\frac{45}{56}$ " 12$\frac{45}{56}$ " " " "	1 bushel.
A " 8$\frac{1}{8}$ × 8$\frac{1}{8}$ " 8$\frac{1}{8}$ " " " "	1 peck.
A " 6$\frac{7}{16}$ × 6$\frac{7}{16}$ " 6$\frac{7}{16}$ " " " "	½ "
A " 4$\frac{1}{16}$ × 4$\frac{1}{16}$ " 4$\frac{1}{16}$ " " " "	1 quart.
An acre contains	4840 square yards.
209 feet long by 209 feet broad is	1 acre.

Table of Distance.

A mile is	5280 feet or 1760 yards.
A knot is	6086 feet.
A league is	3 miles.
A fathom is	6 feet.
A metre is nearly	3 feet 3⅜ inches.
A hand is	4 inches.
A palm is	3 "
A span is	9 "

Measure of Length.

12 inches	1 foot.	4 rods	1 chain.
3 feet	1 yard.	10 chains	1 furlong.
2 yards	1 fathom.	8 furlongs	1 mile.
16½ feet	1 rod.	3 miles	1 league.

Each *nominal* horse-power of boilers requires 1 cubic foot of water per hour.

In calculating horse-power of steam-boilers consider for—
Tubular boilers 15 sq. ft. of heating-surface equivalent to 1 horse-power.
Flue boilers 12 sq. ft. of heating-surface equivalent to 1 horse-power.
Cylinder boilers 10 sq. ft. of heating-surface equivalent to 1 horse-power.

To find the area of a piston, square the diameter and multiply by .7854.

To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by .434.

A horse-power in machinery is estimated at 33,000 pounds raised one foot high in a minute, or one pound raised 33,000 feet high in a minute.

Iron under the influence of the hammer and of constant use gradually assumes, by repeated vibration, a different texture from that it had when the piece was new. The metal becomes crystalline, loses its tenacity, and becomes brittle.

WEIGHT OF WATER.

One cubic foot at 39.1° F. = 62.425 lbs., at 212° F. = 59.833. At 62° F. the weight varies from 62.291 to 62.360. The figure generally believed to be the most accurate is 62.355. Weight of 1 gallon at 39.2° = 8.3389 lbs.

WEIGHTS OF CAST-IRON WATER-PIPES.

IN POUNDS PER FOOT RUN, INCLUDING BELLS AND SPIGOTS.

Diameter.	Philadelphia Water-works.	Chicago Water-works.	Cincinnati.		Regular Standard.	Light.
			Weight.	Thickness.		
2-inch....					7	6
3 "	15.000		17	$\frac{1}{2}$"	15	13
4 "	21.111	24.167	23	"	22	20
6 "	30.106	36.666	50	$\frac{5}{8}$"	33	30
8 "	40.683	50.000	65	"	42	40
10 "	52.075	65.000	80	"	60	55
12 "	69.162	83.333	100	"	75	70
16 "	102.522	125.000	130	"		
20 "	147.681		200	$\frac{7}{8}$"		
24 "		250.000	224	"		
30 "			300	1"		
36 "		450.000	430	$1\frac{1}{8}$"		

Water-pipe is usually tested to 300 pounds pressure per square inch before delivery, and a hammer test should be made while the pipe is under pressure.

The Cincinnati lengths are uniform for all diameters, 12 feet exclusive of bell.

Standard lengths are for 2-inch pipe 8 feet, and all other sizes 12 feet.

THICKNESS OF CAST-IRON WATER-PIPE.

The following formula, adapted from Neville, is believed to be a safe equation for the thickness of cast-iron pipe for public water-supply:

$$t = \frac{9}{S}\left[.0016\left(\frac{h}{33} + 10\right)d\right] + .32,$$

where t = thickness of pipe in inches,

h = head or pressure in feet,

d = diameter of pipe in inches,

S = the tensile strength of metal in tons of 2000 pounds.

What should be the thickness of a 20-inch water-main subject to a maximum pressure of 150 pounds per square inch, or $150 \times 2.308 = 346.2$ feet head, with cast-iron of 18,000 pounds tensile strength?

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 20\right] + .32 = .9757''.$$

What should be the thickness of 40-inch pipe for same service and of same metal?

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 40\right] + .32 = 1.6313''$$

The speed at which millstones should be run is

For 3-feet stones	230 to 250	revolutions	per minute.
" 3½ " "	200	"	" "
" 4 " "	180	"	" "
" 4½ " "	160	"	" "
Speed of bolting-reels	30 to 35	"	" "
" " conveyers for flour . . .	35 to 40	"	" "
" " " " wheat . . .	45 to 50	"	" "
" " elevators	30 to 35	"	" "

" " smut-machines from 550 to 700 revolutions per minute, according to size of machine.

For *merchant* mills allow 20 horse-power to a pair of burrs (4 feet), and the necessary machinery for cleaning and bolting; and for *country* mills about 10 horse-power to a pair of burrs.

For a single upright saw allow 10 horse-power, speed about 150 revolutions per minute.

For circular saws the best average working speed is

650 to 700 rev. per min. for 36-in. saw.	500 to 525 rev. per min. for 48-in. saw.
600 to 650 " " " " 40 " "	475 to 500 " " " " 54 " "
550 to 600 " " " " 42 " "	400 to 450 " " " " 60 " "
525 to 550 " " " " 44 " "	

A 60-saw gin requires 6 horse-power to gin 500 pounds of lint in 2 hours.

A sumac-mill requires 15 horse-power.

To reduce for round cores and core-prints, multiply the square of the diameter by the length of the core in inches, and the product by 0.017 is the weight of the pine core, to be deducted for the weight of the pattern.

SHRINKAGE OF CASTINGS.

Pattern-maker's rule should be for

Cast iron	1/8
Brass	3/16
Lead	1/8
Tin	1/12
Zinc	3/16

of an inch longer per linear foot.

PROPERTIES OF THE CIRCLE.

Diameter × 3.14159 = circumference.
Diameter × .8862 = side of an equal square.
Diameter × .7071 = side of an inscribed square.
Diameter^2 × .7854 = area of circle.
Radius × 6.28318 = circumference.
Circumference ÷ 3.14159 = diameter.

WROUGHT-IRON WELDED TUBES FOR STEAM, GAS, OR WATER.

Nominal Diameter.	Actual Inside Diameter.	Actual Outside Diameter.	Thickness.	Weight per Foot of Length.	No. of Threads per Inch of Screw.
Inches.	Inches.	Inches.	Inches.	Pounds.	
⅛	.270	.405	.068	.243	27
¼	.364	.54	.088	.422	18
⅜	.494	.675	.091	.561	18
½	.623	.84	.109	.845	14
¾	.824	1.05	.113	1.126	14
1	1.048	1.315	.134	1.670	11½
1¼	1.380	1.66	.140	2.258	11½
1½	1.611	1.9	.145	2.694	11½
2	2.067	2.375	.154	3.667	11½
2½	2.468	2.875	.204	5.773	8
3	3.067	3.5	.217	7.547	8
3½	3.548	4.0	.226	9.055	8
4	4.026	4.5	.237	10.728	8
4½	4.508	5.0	.247	12.492	8
5	5.045	5.563	.259	14.564	8
6	6.065	6.625	.280	18.767	8
7	7.023	7.625	.301	23.410	8
8	7.982	8.625	.322	28.348	8
9	9.001	9.688	.344	34.077	8
10	10.019	10.75	.366	40.641	8

DIFFERENT COLORS OF IRON CAUSED BY HEAT. (POUILLET.)

Cent.	Fahr.	Color.
210°	410°	Pale yellow.
221	430	Dull yellow.
256	493	Crimson.
261	502	Violet, purple, and dull blue; between 261° C. and 370° C. it passes to bright blue, to sea-green, and then disappears.
370	680	
500	932	Commences to be covered with a light coating of oxide; loses a good deal of its hardness, becomes much more impressible to the hammer, and can be twisted with ease.
525	977	Becomes nascent red.
700	1292	Sombre red.
800	1472	Nascent cherry.
900	1657	Cherry.
1000	1832	Bright cherry.
1100	2012	Dull orange.
1200	2192	Bright orange.
1300	2372	White.
1400	2552	Brilliant white—welding heat.
1500	2732	Dazzling white.
1600	2912	

TABLE OF DECIMAL EQUIVALENTS OF ONE INCH.

1/64	.015625	17/64	.265625	33/64	.515625	49/64	.765625
1/32	.03125	9/32	.28125	17/32	.53125	25/32	.78125
3/64	.046875	19/64	.296875	35/64	.546875	51/64	.796875
1/16	.0625	5/16	.3125	9/16	.5625	13/16	.8125
5/64	.078125	21/64	.328125	37/64	.578125	53/64	.828125
3/32	.09375	11/32	.34375	19/32	.59375	27/32	.84375
7/64	.109375	23/64	.359375	39/64	.609375	55/64	.859375
1/8	.125	3/8	.375	5/8	.625	7/8	.875
9/64	.140625	25/64	.390625	41/64	.640625	57/64	.890625
5/32	.15625	13/32	.40625	21/32	.65625	29/32	.90625
11/64	.171875	27/64	.421875	43/64	.671875	59/64	.921875
3/16	.1875	7/16	.4375	11/16	.6875	15/16	.9375
13/64	.203125	29/64	.453125	45/64	.703125	61/64	.953125
7/32	.21875	15/32	.46875	23/32	.71875	31/32	.96875
15/64	.234375	31/64	.484375	47/64	.734375	63/64	.984375
1/4	.25	1/2	.50	3/4	.75	1	

MELTING-POINT OF METALS, ETC.

Names.	Fahr.	Names.	Fahr.
Platinum	3227°	Wrought iron	2900°
Antimony	842	Steel, hard	2588
Bismuth	509	Copper	1931
Tin	442	Glass	2012
Lead	617	Beeswax	151
Zinc	779	Sulphur	239
Cast iron	2100	Tallow	92

TABLE 5.

WEIGHT OF VARIOUS SUBSTANCES.

RULE.—Divide the specific gravity of the substance by 16 and the quotient will give the weight of a cubic foot of it in pounds.

Substances—Metals.	Specific Gravity.	Weight of a Cubic Inch.	Substances—Metals.	Specific Gravity.	Weight of a Cubic Inch.
Aluminum	2.560	.0926	Lead, cast	11,352	.4106
Brass, plate	8.380	.3031	Lead, rolled	11,388	.4119
Brass, wire	8.214	.2972	Mercury, + 32°	13,598	.4918
Bronze, gun-metal	8.700	.3147	Mercury, 60°	13,580	.4912
Copper, cast	8.788	.3179	Mercury, 212°	13,370	.4836
Copper, plates	8.698	.3146	Steel, plates	7,806	.2823
Copper, wire	8.880	.3212	Steel, soft	7,833	.2833
Iron, cast	7.207	.2607	Steel, wire	7,847	.2838
Iron, cast, gun-metal	7.308	.264	Tin, Cornish, hammered	7,390	.2673
Iron, wrought bars	7.788	.2817	Zinc, cast	6,861	.2482
Iron, rolled plates	7.704	.2787	Zinc, rolled	7,191	.26

TABLE 6.

WEIGHT OF TIMBER PER CUBIC FOOT.

Ash	46 lbs.	Mahogany, Honduras	35 lbs.
Beech	44 "	" Spanish	53 "
Birch	45 "	Oak, English	54 "
Boxwood	62 "	Pine, red	30 to 44 "
Elm	34 "	" yellow	29 to 41 "
Larch	34 "	" white	30 "
Lignum-vitæ	80 "	Teak	41 to 55 "

CIRCUMFERENCES AND AREAS OF CIRCLES ADVANCING BY EIGHTHS.

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
1/64	.04909	.00019	2 11/16	8.4430	5.6727	6 5/8	20.813	34.472
1/32	.09818	.00077	3/4	8.6394	5.9396	3/4	21.206	35.785
3/64	.14726	.00173	13/16	8.8357	6.2126	7/8	21.598	37.122
1/16	.19635	.00307	7/8	9.0321	6.4918			
3/32	.29452	.00690	15/16	9.2284	6.7771	7	21.991	38.485
1/8	.39270	.01227				1/8	22.384	39.871
5/32	.49087	.01917	3	9.4248	7.0686	1/4	22.776	41.282
3/16	.58905	.02761	1/16	9.6211	7.3662	3/8	23.169	42.718
7/32	.68722	.03758	1/8	9.8175	7.6699	1/2	23.562	44.179
1/4	.78540	.04909	3/16	10.014	7.9798	5/8	23.955	45.664
9/32	.88357	.06213	1/4	10.210	8.2958	3/4	24.347	47.173
5/16	.98175	.07670	5/16	10.407	8.6179	7/8	24.740	48.707
11/32	1.0799	.09281	3/8	10.603	8.9462			
3/8	1.1781	.11045	7/16	10.799	9.2806	8	25.133	50.265
13/32	1.2763	.12962	1/2	10.996	9.6211	1/8	25.525	51.849
7/16	1.3744	.15033	9/16	11.192	9.9678	1/4	25.918	53.456
15/32	1.4726	.17257	5/8	11.388	10.321	3/8	26.311	55.088
1/2	1.5708	.19635	11/16	11.585	10.680	1/2	26.704	56.745
17/32	1.6690	.22166	3/4	11.781	11.045	5/8	27.096	58.426
9/16	1.7671	.24850	13/16	11.977	11.416	3/4	27.489	60.132
19/32	1.8653	.27688	7/8	12.174	11.793	7/8	27.882	61.862
5/8	1.9635	.30680	15/16	12.370	12.177			
21/32	2.0617	.33824				9	28.274	63.617
11/16	2.1598	.37122	4	12.566	12.566	1/8	28.667	65.397
23/32	2.2580	.40574	1/16	12.763	12.962	1/4	29.060	67.201
3/4	2.3562	.44179	1/8	12.959	13.364	3/8	29.452	69.029
25/32	2.4544	.47937	3/16	13.155	13.772	1/2	29.845	70.882
13/16	2.5525	.51849	1/4	13.352	14.186	5/8	30.238	72.760
27/32	2.6507	.55914	5/16	13.548	14.607	3/4	30.631	74.662
7/8	2.7489	.60132	3/8	13.744	15.033	7/8	31.023	76.589
29/32	2.8471	.64504	7/16	13.941	15.466			
15/16	2.9452	.69029	1/2	14.137	15.904	10	31.416	78.540
31/32	3.0434	.73708	9/16	14.334	16.349	1/8	31.809	80.516
			5/8	14.530	16.800	1/4	32.201	82.516
1	3.1416	.7854	11/16	14 726	17.257	3/8	32.594	84.541
1/16	3.3379	.8866	3/4	14.923	17.721	1/2	32.987	86.590
1/8	3.5343	.9940	13/16	15.119	18.190	5/8	33.379	88.664
3/16	3.7306	1.1075	7/8	15.315	18.665	3/4	33.772	90.763
1/4	3.9270	1.2272	15/16	15.512	19.147	7/8	34.165	92.886
5/16	4.1233	1.3530						
3/8	4.3197	1.4849	5	15.708	19.635	11	34.558	95.033
7/16	4.5160	1.6230	1/16	15.904	20.129	1/8	34.950	97.205
1/2	4.7124	1.7671	1/8	16.101	20.629	1/4	35.343	99.402
9/16	4.9087	1.9175	3/16	16.297	21.135	3/8	35.736	101.62
5/8	5.1051	2.0739	1/4	16.493	21.648	1/2	36.128	103.87
11/16	5.3014	2.2365	5/16	16.690	22.166	5/8	36.521	106.14
3/4	5.4978	2.4053	3/8	16.886	22.691	3/4	36.914	108.43
13/16	5.6941	2.5802	7/16	17.082	23.221	7/8	37.306	110.75
7/8	5.8905	2.7612	1/2	17.279	23.758			
15/16	6.0868	2.9483	9/16	17.475	24.301	12	37.699	113.10
			5/8	17.671	24.850	1/8	38.092	115.47
2	6.2832	3.1416	11/16	17.868	25.406	1/4	38.485	117.86
1/16	6.4795	3.3410	3/4	18.064	25.967	3/8	38.877	120.28
1/8	6.6759	3.5466	13/16	18.261	26.535	1/2	39.270	122.72
3/16	6.8722	3.7583	7/8	18.457	27.109	5/8	39.663	125.19
1/4	7.0686	3.9761	15/16	18.653	27.688	3/4	40.055	127.68
5/16	7.2649	4.2000				7/8	40.449	130.19
3/8	7.4613	4.4301	6	18.850	82.274			
7/16	7.6576	4.4664	1/8	19.242	89.465			
1/2	7.8540	4.9087	1/4	19.635	30.680			
9/16	8.0503	5.1572	3/8	20.028	31.919			
5/8	8.2467	5.4119	1/2	20.420	33.183			

To find the weight of castings by the weight of pine patterns, multiply the weight of the pattern by 12 for cast iron, 13 for brass, 19 for lead, 12.2 for tin, 14.4 for zinc, and the product is the weight of the casting.

Number.	Square.	Cube.	Square Root.	Cube Root.
.1	.01	.001	.3162	.4642
.15	.0225	.0034	.3873	.5313
.2	.04	.008	.4472	.5848
.25	.0625	.0156	.500	.6300
.3	.09	.027	.5477	.6694
.35	.1225	.0429	.5916	.7047
.4	.16	.064	.6325	.7368
.45	.2025	.0911	.6708	.7663
.5	.25	.125	.7071	.7937
.55	.3025	.1664	.7416	.8193
.6	.36	.216	.7746	.8434
.65	.4225	.2746	.8062	.8662
.7	.49	.343	.8367	.8879
.75	.5625	.4219	.8660	.9086
.8	.64	.512	.8944	.9283
.85	.7225	.6141	.9219	.9473
.9	.81	.729	.9487	.9655
.95	.9025	.8574	.9747	.9830
1	1	1	1.0000	1.0000
2	4	8	1.4142	1.2599
3	9	27	1.7321	1.4422
4	16	64	2.0000	1.5874
5	25	125	2.2361	1.7100
6	36	216	2.4495	1.8171
7	49	343	2.6458	1.9129
8	64	512	2.8284	2.0000
9	81	729	3.0000	2.0801
10	1 00	1 000	3.1623	2.1544
11	1 21	1 331	3.3166	2.2240
12	1 44	1 728	3.4641	2.2894
13	1 69	2 197	3.6056	2.3513
14	1 96	2 744	3.7417	2.4101
15	2 25	3 375	3.8730	2.4662
16	2 56	4 096	4.0000	2.5198
17	2 89	4 913	4.1231	2.5713
18	3 24	5 832	4.2426	2.6207
19	3 61	6 859	4.3589	2.6684
20	4 00	8 000	4.4721	2.7144
21	4 41	9 261	4.5826	2.7589
22	4 84	10 648	4.6904	2.8020
23	5 29	12 167	4.7958	2.8439
24	5 76	13 824	4.8990	2.8845
25	6 25	15 625	5.0000	2.9240
26	6 76	16 576	5.0990	2.9625
27	7 29	19 683	5.1962	3.0000
28	7 84	21 952	5.2915	3.0366
29	8 41	24 389	5.3852	3.0723
30	9 00	27 000	5.4772	3.1072
31	9 61	29 791	5.5678	3.1414
32	10 24	32 768	5.6569	3.1748
33	10 89	35 937	5.7446	3.2075
34	11 56	39 304	5.8310	3.2396
35	12 25	42 875	5.9161	3.2711
36	12 96	46 656	6.0000	3.3019
37	13 69	50 653	6.0828	3.3322
38	14 44	54 872	6.1644	3.3620
39	15 21	59 319	6.2450	3.3912
40	16 00	64 000	6.3246	3.4200
41	16 81	68 921	6.4031	3.4482
42	17 64	74 088	6.4807	3.4760
43	18 49	79 507	6.5574	3.5034
44	19 36	85 184	6.6332	3.5303
45	20 25	91 125	6.7082	3.5569
46	21 16	97 336	6.7823	3.5830
47	22 09	103 823	6.8557	3.6088
	23 04	110 592	6.9282	3.6342
	24 01	117 649	7.0000	3.6593
	25 00	125 000	7.0711	3.6840

Number	Square.	Cube.	Square Root.	Cube Root.
51	26 01	132 651	7.1414	3.7084
52	27 04	140 608	7.2111	3.7325
53	28 09	148 877	7.2801	3.7563
54	29 16	157 464	7.3485	3.7798
55	30 25	166 375	7.4162	3.8030
56	31 36	175 616	7.4833	3.8259
57	32 49	185 193	7.5498	3.8485
58	33 64	195 112	7.6158	3.8709
59	34 81	205 379	7.6811	3.8930
60	36 00	216 000	7.7460	3.9149
61	37 21	226 981	7.8102	3.9365
62	38 44	238 328	7.8740	3.9579
63	39 69	250 047	7.9373	3.9791
64	40 96	262 144	8.0000	4.0000
65	42 25	274 625	8.0623	4.0207
66	43 56	287 496	8.1240	4.0412
67	44 89	300 763	8.1854	4.0615
68	46 24	314 432	8.2462	4.0817
69	47 61	328 509	8.3066	4.1016
70	49 00	343 000	8.3666	4.1213
71	50 41	357 911	8.4261	4.1408
72	51 84	373 248	8.4853	4.1602
73	53 29	389 017	8.5440	4.1793
74	54 76	405 224	8.6023	4.1983
75	56 25	421 875	8.6603	4.2172
76	57 76	438 976	8.7178	4.2358
77	59 29	456 533	8.7750	4.2543
78	60 84	474 552	8.8318	4.2727
79	62 41	493 039	8.8882	4.2908
80	64 00	512 000	8.9443	4.3089
81	65 61	531 441	9.0000	4.3267
82	67 24	551 368	9.0554	4.3445
83	68 89	571 787	9.1104	4.3621
84	70 56	592 704	9.1652	4.3795
85	72 25	614 125	9.2195	4.3968
86	73 96	636 056	9.2736	4.4140
87	75 69	658 503	9.3274	4.4310
88	77 44	681 472	9.3808	4.4480
89	79 21	704 969	9.4340	4.4647
90	81 00	729 000	9.4868	4.4814
91	82 81	753 571	9.5394	4.4979
92	84 64	778 688	9.5917	4.5144
93	86 49	804 357	9.6437	4.5307
94	88 36	830 584	9.6954	4.5468
95	90 25	857 375	9.7468	4.5629
96	92 16	884 736	9.7980	4.5789
97	94 09	912 673	9.8489	4.5947
98	96 04	941 192	9.8995	4.6104
99	98 01	970 299	9.9499	4.6261
100	1 00 00	1 000 000	10.0000	4.6416

Number	Square.	Cube.	Square Root.	Cube Root.
101	1 02 01	1 030 301	10.0499	4.6570
102	1 04 04	1 061 208	10.0995	4.6723
103	1 06 09	1 092 727	10.1489	4.6875
104	1 08 16	1 124 864	10.1980	4.7027
105	1 10 25	1 157 625	10.2470	4.7177
106	1 12 36	1 191 016	10.2956	4.7326
107	1 14 49	1 225 043	10.3441	4.7475
108	1 16 64	1 259 712	10.3923	4.7622
109	1 18 81	1 295 029	10.4403	4.7769
110	1 21 00	1 331 000	10.4881	4.7914
111	1 23 21	1 367 631	10.5357	4.8059
112	1 25 44	1 404 928	10.5830	4.8203
113	1 27 69	1 442 897	10.6301	4.8346
114	1 29 96	1 481 544	10.6771	4.8488
115	1 32 25	1 520 875	10.7238	4.8629
116	1 34 56	1 560 896	10.7703	4.8770
117	1 36 89	1 601 613	10.8167	4.8910
118	1 39 24	1 643 032	10.8628	4.9049
119	1 41 61	1 685 159	10.9087	4.9187
120	1 44 00	1 728 000	10.9545	4.9324
121	1 46 41	1 771 561	11.0000	4.9461
122	1 48 84	1 815 848	11.0454	4.9597
123	1 51 29	1 860 867	11.0905	4.9732
124	1 53 76	1 906 624	11.1355	4.9866
125	1 56 25	1 953 125	11.1803	5.0000
126	1 58 76	2 000 376	11.2250	5.0133
127	1 61 29	2 048 383	11.2694	5.0265
128	1 63 84	2 097 152	11.3137	5.0397
129	1 66 41	2 146 689	11.3578	5.0528
130	1 69 00	2 197 000	11.4018	5.0658
131	1 71 61	2 248 091	11.4455	5.0788
132	1 74 24	2 299 968	11.4891	5.0916
133	1 76 89	2 352 637	11.5326	5.1045
134	1 79 56	2 406 104	11.5758	5.1172
135	1 82 25	2 460 375	11.6190	5.1299
136	1 84 96	2 515 456	11.6619	5.1426
137	1 87 69	2 571 353	11.7047	5.1551
138	1 90 44	2 628 072	11.7473	5.1676
139	1 93 21	2 685 619	11.7898	5.1801
140	1 96 00	2 744 000	11.8322	5.1925
141	1 98 81	2 803 221	11.8743	5.2048
142	2 01 64	2 863 288	11.9164	5.2171
143	2 04 49	2 924 207	11.9583	5.2293
144	2 07 36	2 985 984	12.0000	5.2415
145	2 10 25	3 048 625	12.0416	5.2536
146	2 13 16	3 112 136	12.0830	5.2656
147	2 16 09	3 176 523	12.1244	5.2776
148	2 19 04	3 241 792	12.1655	5.2896
149	2 22 01	3 307 949	12.2066	5.3015
150	2 25 00	3 375 000	12.2474	5.3133

Number	Square.	Cube.	Square Root.	Cube Root.
151	2 28 01	3 442 951	12.2882	5.3251
152	2 31 04	3 511 808	12.3288	5.3368
153	2 34 09	3 581 577	12.3693	5.3485
154	2 37 16	3 652 264	12.4097	5.3601
155	2 40 25	3 723 875	12.4499	5.3717
156	2 43 36	3 796 416	12.4900	5.3832
157	2 46 49	3 869 893	12.5300	5.3947
158	2 49 64	3 944 312	12.5698	5.4061
159	2 52 81	4 019 679	12.6095	5.4175
160	2 56 00	4 096 000	12.6491	5.4288
161	2 59 21	4 173 281	12.6886	5.4401
162	2 62 44	4 251 528	12.7279	5.4514
163	2 65 69	4 330 747	12.7671	5.4626
164	2 68 96	4 410 944	12.8062	5.4737
165	2 72 25	4 492 125	12.8452	5.4848
166	2 75 56	4 574 296	12.8841	5.4959
167	2 78 89	4 657 463	12.9228	5.5069
168	2 82 24	4 741 632	12.9615	5.5178
169	2 85 61	4 826 809	13.0000	5.5288
170	2 89 00	4 913 000	13.0384	5.5397
171	2 92 41	5 000 211	13.0767	5.5505
172	2 95 84	5 088 448	13.1149	5.5613
173	2 99 29	5 177 717	13.1529	5.5721
174	3 02 76	5 268 024	13.1909	5.5828
175	3 06 25	5 359 375	13.2288	5.5934
176	3 09 76	5 451 776	13.2665	5.6041
177	3 13 29	5 545 233	13.3041	5.6147
178	3 16 84	5 639 752	13.3417	5.6252
179	3 20 41	5 735 339	13.3791	5.6357
180	3 24 00	5 832 000	13.4164	5.6462
181	3 27 61	5 929 741	13.4536	5.6567
182	3 31 24	6 028 568	13.4907	5.6671
183	3 34 89	6 128 487	13.5277	5.6774
184	3 38 56	6 229 504	13.5647	5.6877
185	3 42 25	6 331 625	13.6015	5.6980
186	3 45 96	6 434 856	13.6382	5.7083
187	3 49 69	6 539 203	13.6748	5.7185
188	3 53 44	6 644 672	13.7113	5.7287
189	3 57 21	6 751 269	13.7477	5.7388
190	3 61 00	6 859 000	13.7840	5.7489
191	3 64 81	6 967 871	13.8203	5.7590
192	3 68 64	7 077 888	13.8564	5.7690
193	3 72 49	7 189 057	13.8924	5.7790
194	3 76 36	7 301 384	13.9284	5.7890
195	3 80 25	7 414 875	13.9642	5.7989
196	3 84 16	7 529 536	14.0000	5.8088
197	3 88 09	7 645 373	14.0357	5.8186
198	3 92 04	7 762 392	14.0712	5.8285
199	3 96 01	7 880 599	14.1067	5.8383
200	4 00 00	8 000 000	14.1421	5.8480

Number	Square.	Cube.	Square Root.	Cube Root.
201	4 04 01	8 120 601	14.1774	5.8578
202	4 08 04	8 242 408	14.2127	5.8675
203	4 12 09	8 365 427	14.2478	5.8771
204	4 16 16	8 489 664	14.2829	5.8868
205	4 20 25	8 615 125	14.3178	5.8964
206	4 24 36	8 741 816	14.3527	5.9059
207	4 28 49	8 869 743	14.3875	5.9155
208	4 32 64	8 998 912	14.4222	5.9250
209	4 36 81	9 129 329	14.4568	5.9345
210	4 41 00	9 261 000	14.4914	5.9439
211	4 45 21	9 393 931	14.5258	5.9533
212	4 49 44	9 528 128	14.5602	5.9627
213	4 53 69	9 663 597	14.5945	5.9721
214	4 57 96	9 800 344	14.6287	5.9814
215	4 62 25	9 938 375	14.6629	5.9907
216	4 66 56	10 077 696	14.6969	6.0000
217	4 70 89	10 218 313	14.7309	6.0092
218	4 75 24	10 360 232	14.7648	6.0185
219	4 79 61	10 503 459	14.7986	6.0277
220	4 84 00	10 648 000	14.8324	6.0368
221	4 88 41	10 793 861	14.8661	6.0459
222	4 92 84	10 941 048	14.8997	6.0550
223	4 97 29	11 089 567	14.9332	6.0641
224	5 01 76	11 239 424	14.9666	6.0732
225	5 06 25	11 390 625	15.0000	6.0822
226	5 10 76	11 543 176	15.0333	6.0912
227	5 15 29	11 697 083	15.0665	6.1002
228	5 19 84	11 852 352	15.0997	6.1091
229	5 24 41	12 008 989	15.1327	6.1180
230	5 29 00	12 167 000	15.1658	6.1269
231	5 33 61	12 326 391	15.1987	6.1358
232	5 38 24	12 487 168	15.2315	6.1446
233	5 42 89	12 649 337	15.2643	6.1534
234	5 47 56	12 812 904	15.2971	6.1622
235	5 52 25	12 977 875	15.3297	6.1710
236	5 56 96	13 144 256	15.3623	6.1797
237	5 61 69	13 312 053	15.3948	6.1885
238	5 66 44	13 481 272	15.4272	6.1972
239	5 71 21	13 651 919	15.4596	6.2058
240	5 76 00	13 824 000	15.4919	6.2145
241	5 80 81	13 997 521	15.5242	6.2231
242	5 85 64	14 172 488	15.5563	6.2317
243	5 90 49	14 348 907	15.5885	6.2403
244	5 95 36	14 526 784	15.6205	6.2488
245	6 00 25	14 706 125	15.6525	6.2573
246	6 05 16	14 886 936	15.6844	6.2658
247	6 10 09	15 069 223	15.7162	6.2743
248	6 15 04	15 252 992	15.7480	6.2828
249	6 20 01	15 438 249	15.7797	6.2912
250	6 25 00	15 625 000	15.8114	6.2996

Number	Square.	Cube.	Square Root.	Cube Root.
251	6 30 01	15 813 251	15.8430	6.3080
252	6 35 04	16 003 008	15.8745	6.3164
253	6 40 09	16 194 277	15.9060	6.3247
254	6 45 16	16 387 064	15.9374	6.3330
255	6 50 25	16 581 375	15.9687	6.3413
256	6 55 36	16 777 216	16.0000	6.3496
257	6 60 49	16 974 593	16.0312	6.3579
258	6 65 64	17 173 512	16.0624	6.3661
259	6 70 81	17 373 979	16.0935	6.3743
260	6 76 00	17 576 000	16.1245	6.3825
261	6 81 21	17 779 581	16.1555	6.3907
262	6 86 44	17 984 728	16.1864	6.3988
263	6 91 69	18 191 447	16.2173	6.4070
264	6 96 96	18 399 744	16.2481	6.4151
265	7 02 25	18 609 625	16.2788	6.4232
266	7 07 56	18 821 096	16.3095	6.4312
267	7 12 89	19 034 163	16.3401	6.4393
268	7 18 24	19 248 832	16.3707	6.4473
269	7 23 61	19 465 109	16.4012	6.4553
270	7 29 00	19 683 000	16.4317	6.4633
271	7 34 41	19 902 511	16.4621	6.4713
272	7 39 84	20 123 648	16.4924	6.4792
273	7 45 29	20 346 417	16.5227	6.4872
274	7 50 76	20 570 824	16.5529	6.4951
275	7 56 25	20 796 875	16.5831	6.5030
276	7 61 76	21 024 576	16.6132	6.5108
277	7 67 29	21 253 933	16.6433	6.5187
278	7 72 84	21 484 952	16.6733	6.5265
279	7 78 41	21 717 639	16.7033	6.5343
280	7 84 00	21 952 000	16.7332	6.5421
281	7 89 61	22 188 041	16.7631	6.5499
282	7 95 24	22 425 768	16.7929	6.5577
283	8 00 89	22 665 187	16.8226	6.5654
284	8 06 56	22 906 304	16.8523	6.5731
285	8 12 25	23 149 125	16.8819	6.5808
286	8 17 96	23 393 656	16.9115	6.5885
287	8 23 69	23 639 903	16.9411	6.5962
288	8 29 44	23 887 872	16.9706	6.6039
289	8 35 21	24 137 569	17.0000	6.6115
290	8 41 00	24 389 000	17.0294	6.6191
291	8 46 81	24 642 171	17.0587	6.6267
292	8 52 64	24 897 088	17.0880	6.6343
293	8 58 49	25 153 757	17.1172	6.6419
294	8 64 36	25 412 184	17.1464	6.6494
295	8 70 25	25 672 375	17.1756	6.6569
296	8 76 16	25 934 336	17.2047	6.6644
297	8 82 09	26 198 073	17.2337	6.6719
298	8 88 04	26 463 592	17.2627	6.6794
299	8 94 01	26 730 899	17.2916	6.6869
300	9 00 00	27 000 000	17.3205	6.6943

Number	Square.	Cube.	Square Root.	Cube Root.
301	9 06 01	27 270 901	17.3494	6.7018
302	9 12 04	27 543 608	17.3781	6.7092
303	9 18 09	27 818 127	17.4069	6.7166
304	9 24 16	28 094 464	17.4356	6.7240
305	9 30 25	28 372 625	17.4642	6.7313
306	9 36 36	28 652 616	17.4929	6.7387
307	9 42 49	28 934 443	17.5214	6.7460
308	9 48 64	29 218 112	17.5499	6.7533
309	9 54 81	29 503 629	17.5784	6.7606
310	9 61 00	29 791 000	17.6068	6.7679
311	9 67 21	30 080 231	17.6352	6.7752
312	9 73 44	30 371 328	17.6635	6.7824
313	9 79 69	30 664 297	17.6918	6.7897
314	9 85 96	30 959 144	17.7200	6.7969
315	9 92 25	31 255 875	17.7482	6.8041
316	9 98 56	31 554 496	17.7764	6.8113
317	10 04 89	31 855 013	17.8045	6.8185
318	10 11 24	32 157 432	17.8326	6.8256
319	10 17 61	32 461 759	17.8606	6.8328
320	10 24 00	32 768 000	17.8885	6.8399
321	10 30 41	33 076 161	17.9165	6.8470
322	10 36 84	33 386 248	17.9444	6.8541
323	10 43 29	33 698 267	17.9722	6.8612
324	10 49 76	34 012 224	18.0000	6.8683
325	10 56 25	34 328 125	18.0278	6.8753
326	10 62 76	34 645 976	18.0555	6.8824
327	10 69 29	34 965 783	18.0831	6.8894
328	10 75 84	35 287 552	18.1108	6.8964
329	10 82 41	35 611 289	18.1384	6.9034
330	10 89 00	35 937 000	18.1659	6.9104
331	10 95 61	36 264 691	18.1934	6.9174
332	11 02 24	36 594 368	18.2209	6.9244
333	11 08 89	36 926 037	18.2483	6.9313
334	11 15 56	37 259 704	18.2757	6.9382
335	11 22 25	37 595 375	18.3030	6.9451
336	11 28 96	37 933 056	18.3303	6.9521
337	11 35 69	38 272 753	18.3576	6.9589
338	11 42 44	38 614 472	18.3848	6.9658
339	11 49 21	38 958 219	18.4120	6.9727
340	11 56 00	39 304 000	18.4391	6.9795
341	11 62 81	39 651 821	18.4662	6.9864
342	11 69 64	40 001 688	18.4932	6.9932
343	11 76 49	40 353 607	18.5203	7.0000
344	11 83 36	40 707 584	18.5472	7.0068
345	11 90 25	41 063 625	18.5742	7.0136
346	11 97 16	41 421 736	18.6011	7.0203
347	12 04 09	41 781 923	18.6279	7.0271
348	12 11 04	42 144 192	18.6548	7.0338
349	12 18 01	42 508 549	18.6815	7.0406
350	12 25 00	42 875 000	18.7083	7.0473

Number	Square.	Cube.	Square Root.	Cube Root.
351	12 32 01	43 243 551	18.7350	7.0540
352	12 39 04	43 614 208	18.7617	7.0607
353	12 46 09	43 986 977	18.7883	7.0674
354	12 53 16	44 361 864	18.8149	7.0740
355	12 60 25	44 738 875	18.8414	7.0807
356	12 67 36	45 118 016	18.8680	7.0873
357	12 74 49	45 499 293	18.8944	7.0940
358	12 81 64	45 882 712	18.9209	7.1006
359	12 88 81	46 268 279	18.9473	7.1072
360	12 96 00	46 656 000	18.9737	7.1138
361	13 03 21	47 045 881	19 0000	7.1204
362	13 10 44	47 437 928	19.0263	7.1269
363	13 17 69	47 832 147	19.0526	7.1335
364	13 24 96	48 228 544	19.0788	7.1400
365	13 32 25	48 627 125	19.1050	7.1466
366	13 39 56	49 027 896	19.1311	7.1531
367	13 46 89	49 430 863	19.1572	7.1596
368	13 54 24	49 836 032	19.1833	7.1661
369	13 61 61	50 243 409	19.2094	7.1726
370	13 69 00	50 653 000	19.2354	7.1791
371	13 76 41	51 064 811	19.2614	7.1855
372	13 83 84	51 478 848	19.2873	7.1920
373	13 91 29	51 895 117	19.3132	7.1984
374	13 98 76	52 313 624	19.3391	7.2048
375	14 06 25	52 734 375	19.3649	7.2112
376	14 13 76	53 157 376	19.3907	7.2177
377	14 21 29	53 582 633	19.4165	7.2240
378	14 28 84	54 010 152	19.4422	7.2304
379	14 36 41	54 439 939	19.4679	7.2368
380	14 44 00	54 872 000	19.4936	7.2432
381	14 51 61	55 306 341	19.5192	7.2495
382	14 59 24	55 742 968	19.5448	7.2558
383	14 66 89	56 181 887	19.5704	7.2622
384	14 74 56	56 623 104	19.5959	7.2685
385	14 82 25	57 066 625	19.6214	7.2748
386	14 89 96	57 512 456	19.6469	7.2811
387	14 97 69	57 960 603	19.6723	7.2874
388	15 05 44	58 411 072	19.6977	7.2936
389	15 13 21	58 863 869	19.7231	7.2999
390	15 21 00	59 319 000	19.7484	7.3061
391	15 28 81	59 776 471	19.7737	7.3124
392	15 36 64	60 236 288	19.7990	7.3186
393	15 44 49	60 698 457	19.8242	7.3248
394	15 52 36	61 162 984	19.8494	7.3310
395	15 60 25	61 629 875	19.8746	7.3372
396	15 68 16	62 099 136	19.8997	7.3434
397	15 76 09	62 570 773	19.9249	7.3496
398	15 84 04	63 044 792	19.9499	7.3558
399	15 92 01	63 521 199	19.9750	7.3619
400	16 00 00	64 000 000	20.0000	7.3681

Number	Square.	Cube.	Square Root.	Cube Root.
401	16 08 01	64 481 201	20.0250	7.3742
402	16 16 04	64 964 808	20.0499	7.3803
403	16 24 09	65 450 827	20.0749	7.3864
404	16 32 16	65 939 264	20.0998	7.3925
405	16 40 25	66 430 125	20.1246	7.3986
406	16 48 36	66 923 416	20.1494	7.4047
407	16 56 49	67 419 143	20.1742	7.4108
408	16 64 64	67 917 312	20.1990	7.4169
409	16 72 81	68 417 929	20.2237	7.4229
410	16 81 00	68 921 000	20.2485	7.4290
411	16 89 21	69 426 531	20.2731	7.4350
412	16 97 44	69 934 528	20.2978	7.4410
413	17 05 69	70 444 997	20.3224	7.4470
414	17 13 96	70 957 944	20.3470	7.4530
415	17 22 25	71 473 375	20.3715	7.4590
416	17 30 56	71 991 296	20.3961	7.4650
417	17 38 89	72 511 713	20.4206	7.4710
418	17 47 24	73 034 632	20.4450	7.4770
419	17 55 61	73 560 059	20.4695	7.4829
420	17 64 00	74 088 000	20.4939	7.4889
421	17 72 41	74 618 461	20.5183	7.4948
422	17 80 84	75 151 448	20.5426	7.5007
423	17 89 29	75 686 967	20.5670	7.5067
424	17 97 76	76 225 024	20.5913	7.5126
425	18 06 25	76 765 625	20.6155	7.5185
426	18 14 76	77 308 776	20.6398	7.5244
427	18 23 29	77 854 483	20.6640	7.5302
428	18 31 84	78 402 752	20.6882	7.5361
429	18 40 41	78 953 589	20.7123	7.5420
430	18 49 00	79 507 000	20.7364	7.5478
431	18 57 61	80 062 991	20.7605	7.5537
432	18 66 24	80 621 568	20.7846	7.5595
433	18 74 89	81 182 737	20.8087	7.5654
434	18 83 56	81 746 504	20.8327	7.5712
435	18 92 25	82 312 875	20.8567	7.5770
436	19 00 96	82 881 856	20.8806	7.5828
437	19 09 69	83 453 453	20.9045	7.5886
438	19 18 44	84 027 672	20.9284	7.5944
439	19 27 21	84 604 519	20.9523	7.6001
440	19 36 00	85 184 000	20.9762	7.6059
441	19 44 81	85 766 121	21.0000	7.6117
442	19 53 64	86 350 888	21.0238	7.6174
443	19 62 49	86 938 307	21.0476	7.6232
444	19 71 36	87 528 384	21.0713	7.6289
445	19 80 25	88 121 125	21.0950	7.6346
446	19 89 16	88 716 536	21.1187	7.6403
447	19 98 09	89 314 623	21.1424	7.6460
448	20 07 04	89 915 392	21.1660	7.6517
449	20 16 01	90 518 849	21.1896	7.6574
450	20 25 00	91 125 000	21.2132	7.6631

Number	Square.	Cube.	Square Root.	Cube Root.
451	20 34 01	91 733 851	21.2368	7.6688
452	20 43 04	92 345 408	21.2603	7.6744
453	20 52 09	92 959 677	21.2838	7.6801
454	20 61 16	93 576 664	21.3073	7.6857
455	20 70 25	94 196 375	21.3307	7.6914
456	20 79 36	94 818 816	21.3542	7.6970
457	20 88 49	95 443 993	21.3776	7.7026
458	20 97 64	96 071 912	21.4009	7.7082
459	21 06 81	96 702 579	21.4243	7.7138
460	21 16 00	97 336 000	21.4476	7.7194
461	21 25 21	97 972 181	21.4709	7.7250
462	21 34 44	98 611 128	21.4942	7.7306
463	21 43 69	99 252 847	21.5174	7.7362
464	21 52 96	99 897 344	21.5407	7.7418
465	21 62 25	100 544 625	21.5639	7.7473
466	21 71 56	101 194 696	21.5870	7.7529
467	21 80 89	101 847 563	21.6102	7.7584
468	21 90 24	102 503 232	21.6333	7.7639
469	21 99 61	103 161 709	21.6564	7.7695
470	22 09 00	103 823 000	21.6795	7.7750
471	22 18 41	104 487 111	21.7025	7.7805
472	22 27 84	105 154 048	21.7256	7.7860
473	22 37 29	105 823 817	21.7486	7.7915
474	22 46 76	106 496 424	21.7715	7.7970
475	22 56 25	107 171 875	21.7945	7.8025
476	22 65 76	107 850 176	21.8174	7.8079
477	22 75 29	108 531 333	21.8403	7.8134
478	22 84 84	109 215 352	21.8632	7.8188
479	22 94 41	109 902 239	21.8861	7.8243
480	23 04 00	110 592 000	21.9089	7.8297
481	23 13 61	111 284 641	21.9317	7.8352
482	23 23 24	111 980 168	21.9545	7.8406
483	23 32 89	112 678 587	21.9773	7.8460
484	23 42 56	113 379 904	22.0000	7.8514
485	23 52 25	114 084 125	22.0227	7.8568
486	23 61 96	114 791 256	22.0454	7.8622
487	23 71 69	115 501 303	22.0681	7.8676
488	23 81 44	116 214 272	22.0907	7.8730
489	23 91 21	116 930 169	22.1133	7.8784
490	24 01 00	117 649 000	22.1359	7.8837
491	24 10 81	118 370 771	22.1585	7.8891
492	24 20 64	119 095 488	22.1811	7.8944
493	24 30 49	119 823 157	22.2036	7.8998
494	24 40 36	120 553 784	22.2261	7.9051
495	24 50 25	121 287 375	22.2486	7.9105
496	24 60 16	122 023 936	22.2711	7.9158
497	24 70 09	122 763 473	22.2935	7.9211
498	24 80 04	123 505 992	22.3159	7.9264
499	24 90 01	124 251 499	22.3383	7.9317
500	25 00 00	125 000 000	22.3607	7.9370

Number	Square.	Cube.	Square Root.	Cube Root.
501	25 10 01	125 751 501	22.3830	7.9423
502	25 20 04	126 506 008	22.4054	7.9476
503	25 30 09	127 263 527	22.4277	7.9528
504	25 40 16	128 024 064	22.4499	7.9581
505	25 50 25	128 787 625	22.4722	7.9634
506	25 60 36	129 554 216	22.4944	7.9686
507	25 70 49	130 323 843	22.5167	7.9739
508	25 80 63	131 096 512	22.5389	7.9791
509	25 90 81	131 872 229	22.5610	7.9843
510	26 01 00	132 651 000	22.5832	7.9896
511	26 11 21	133 432 831	22.6053	7.9948
512	26 21 44	134 217 728	22.6274	8.0000
513	26 31 69	135 005 697	22.6495	8.0052
514	26 41 96	135 796 744	22.6716	8.0104
515	26 52 25	136 590 875	22.6936	8.0156
516	26 62 56	137 388 096	22.7156	8.0208
517	26 72 89	138 188 413	22.7376	8.0260
518	26 83 24	138 991 832	22.7596	8.0311
519	26 93 61	139 798 359	22.7816	8.0363
520	27 04 00	140 608 000	22.8035	8.0415
521	27 14 41	141 420 761	22.8254	8.0466
522	27 24 84	142 236 648	22.8473	8.0517
523	27 35 29	143 055 667	22.8692	8.0569
524	27 45 76	143 877 824	22.8910	8.0620
525	27 56 25	144 703 125	22.9129	8.0671
526	27 66 76	145 531 576	22.9347	8.0723
527	27 77 29	146 363 183	22.9565	8.0774
528	27 87 84	147 197 952	22.9783	8.0825
529	27 98 41	148 035 889	23.0000	8.0876
530	28 09 00	148 877 000	23.0217	8.0927
531	28 19 61	149 721 291	23.0434	8.0978
532	28 30 24	150 568 768	23.0651	8.1028
533	28 40 89	151 419 437	23.0868	8.1079
534	28 51 56	152 273 304	23.1084	8.1130
535	28 62 25	153 130 375	23.1301	8.1180
536	28 72 96	153 990 656	23.1517	8.1231
537	28 83 69	154 854 153	23.1733	8.1281
538	28 94 44	155 720 872	23.1948	8.1332
539	29 05 21	156 590 819	23.2164	8.1382
540	29 16 00	157 464 000	23.2379	8.1433
541	29 26 81	158 340 421	23.2594	8.1483
542	29 37 64	159 220 088	23.2809	8.1533
543	29 48 49	160 103 007	23.3024	8.1583
544	29 59 36	160 989 184	23.3238	8.1633
545	29 70 25	161 878 625	23.3452	8.1683
546	29 81 16	162 771 336	23.3666	8.1733
547	29 92 09	163 667 323	23.3880	8.1783
548	30 03 04	164 566 592	23.4094	8.1833
549	30 14 01	165 469 149	23.4307	8.1882
550	30 25 00	166 375 000	23.4521	8.1932

Number	Square.	Cube.	Square Root.	Cube Root.
551	30 36 01	167 284 151	23.4734	8.1982
552	30 47 04	168 196 608	23.4947	8.2031
553	30 58 09	169 112 377	23.5160	8.2081
554	30 69 16	170 031 464	23.5372	8.2130
555	30 80 25	170 953 875	23.5584	8.2180
556	30 91 36	171 879 616	23.5797	8.2229
557	31 02 49	172 808 693	23.6008	8.2278
558	31 13 64	173 741 112	23.6220	8.2327
559	31 24 81	174 676 879	23.6432	8.2377
560	31 36 00	175 616 000	23.6643	8.2426
561	31 47 21	176 558 481	23.6854	8.2475
562	31 58 44	177 504 328	23.7065	8.2524
563	31 69 69	178 453 547	23.7276	8.2573
564	31 80 96	179 406 144	23.7487	8.2621
565	31 92 25	180 362 125	23.7697	8.2670
566	32 03 56	181 321 496	23.7908	8.2719
567	32 14 89	182 284 263	23.8118	8.2768
568	32 26 24	183 250 432	23.8328	8.2816
569	32 37 61	184 220 009	23.8537	8.2865
570	32 49 00	185 193 000	23.8747	8.2913
571	32 60 41	186 169 411	23.8956	8.2962
572	32 71 84	187 149 248	23.9165	8.3010
573	32 83 29	188 132 517	23.9374	8.3059
574	32 94 76	189 119 224	23.9583	8.3107
575	33 06 25	190 109 375	23.9792	8.3155
576	33 17 76	191 102 976	24.0000	8.3203
577	33 29 29	192 100 033	24.0208	8.3251
578	33 40 84	193 100 552	24.0416	8.3300
579	33 52 41	194 104 539	24.0624	8.3348
580	33 64 00	195 112 000	24.0832	8.3396
581	33 75 61	196 122 941	24.1039	8.3443
582	33 87 24	197 137 368	24.1247	8.3491
583	33 98 89	198 155 287	24.1454	8.3539
584	34 10 56	199 176 704	24.1661	8.3587
585	34 22 25	200 201 625	24.1868	8.3634
586	34 33 96	201 230 056	24.2074	8.3682
587	34 45 69	202 262 003	24.2281	8.3730
588	34 57 44	203 297 472	24.2487	8.3777
589	34 69 21	204 336 469	24.2693	8.3825
590	34 81 00	205 379 000	24.2899	8.3872
591	34 92 81	206 425 071	24.3105	8.3919
592	35 04 64	207 474 688	24.3311	8.3967
593	35 16 49	208 527 857	24.3516	8.4014
594	35 28 36	209 584 584	24.3721	8.4061
595	35 40 25	210 644 875	24.3926	8.4108
596	35 52 16	211 708 736	24.4131	8.4155
597	35 64 09	212 776 173	24.4336	8.4202
598	35 76 04	213 847 192	24.4540	8.4249
599	35 88 01	214 921 799	24.4745	8.4296
600	36 00 00	216 000 000	24.4949	8.4343

Number	Square.	Cube.	Square Root.	Cube Root.
601	36 12 01	217 081 801	24.5153	8.4390
602	36 24 04	218 167 208	24.5357	8.4437
603	36 36 09	219 256 227	24.5561	8.4484
604	36 48 16	220 348 864	24.5764	8.4530
605	36 60 25	221 445 125	24.5967	8.4577
606	36 72 36	222 545 016	24.6171	8.4623
607	36 84 49	223 648 543	24.6374	8.4670
608	36 96 64	224 755 712	24.6577	8.4716
609	37 08 81	225 866 529	24.6779	8.4763
610	37 21 00	226 981 000	24.6982	8.4809
611	37 33 21	228 099 131	24.7184	8.4856
612	37 45 44	229 220 928	24.7386	8.4902
613	37 57 69	230 346 397	24.7588	8.4948
614	37 69 96	231 475 544	24.7790	8.4994
615	37 82 25	232 608 375	24.7992	8.5040
616	37 94 56	233 744 896	24.8193	8.5086
617	38 06 89	234 885 113	24.8395	8.5132
618	38 19 24	236 029 032	24.8596	8.5178
619	38 31 61	237 176 659	24.8797	8.5224
620	38 44 00	238 328 000	24.8998	8.5270
621	38 56 41	239 483 061	24.9199	8.5316
622	38 68 84	240 641 848	24.9399	8.5362
623	38 81 29	241 804 367	24.9600	8.5408
624	38 93 76	242 970 624	24.9800	8.5453
625	39 06 25	244 140 625	25.0000	8.5499
626	39 18 76	245 314 376	25.0200	8.5544
627	39 31 29	246 491 883	25.0400	8.5590
628	39 43 84	247 673 152	25.0599	8.5635
629	39 56 41	248 858 189	25.0799	8.5681
630	39 69 00	250 047 000	25.0998	8.5726
631	39 81 61	251 239 591	25.1197	8.5772
632	39 94 24	252 435 968	25.1396	8.5817
633	40 06 89	253 636 137	25.1595	8.5862
634	40 19 56	254 840 104	25.1794	8.5907
635	40 32 25	256 047 875	25.1992	8.5952
636	40 44 96	257 259 456	25.2190	8.5997
637	40 57 69	258 474 853	25.2389	8.6043
638	40 70 44	259 694 072	25.2587	8.6088
639	40 83 21	260 917 119	25.2784	8.6132
640	40 96 00	262 144 000	25.2982	8.6177
641	41 08 81	263 374 721	25.3180	8.6222
642	41 21 64	264 609 288	25.3377	8.6267
643	41 34 49	265 847 707	25.3574	8.6312
644	41 47 36	267 089 984	25.3772	8.6357
645	41 60 25	268 336 125	25.3969	8.6401
646	41 73 16	269 586 136	25.4165	8.6446
647	41 86 09	270 840 023	25.4362	8.6490
648	41 99 04	272 097 792	25.4558	8.6535
649	42 12 01	273 359 549	25.4755	8.6579
650	42 25 00	274 625 000	25.4951	8.6624

Number.	Square.	Cube.	Square Root.	Cube Root.
651	42 38 01	275 894 451	25.5147	8.6668
652	42 51 04	277 167 808	25.5343	8.6713
653	42 64 09	278 445 077	25.5539	8.6757
654	42 77 16	279 726 264	25.5734	8.6801
655	42 90 25	281 011 375	25.5930	8.6845
656	43 03 36	282 300 416	25.6125	8.6890
657	43 16 49	283 593 393	25.6320	8.6934
658	43 29 64	284 890 312	25.6515	8.6978
659	43 42 81	286 191 179	25.6710	8.7022
660	43 56 00	287 496 000	25.6905	8.7066
661	43 69 21	288 804 781	25.7099	8.7110
662	43 82 44	290 117 528	25.7294	8.7154
663	43 95 69	291 434 247	25.7488	8.7198
664	44 08 96	292 754 944	25.7682	8.7241
665	44 22 25	294 079 625	25.7876	8.7285
666	44 35 56	295 408 296	25.8070	8.7329
667	44 48 89	296 740 963	25.8263	8.7373
668	44 62 24	298 077 632	25.8457	8.7416
669	44 75 61	299 418 309	25.8650	8.7460
670	44 89 00	300 763 000	25.8844	8.7503
671	45 02 41	302 111 711	25.9037	8.7547
672	45 15 84	303 464 448	25.9230	8.7590
673	45 29 29	304 821 217	25.9422	8.7634
674	45 42 76	306 182 024	25.9615	8.7677
675	45 56 25	307 546 875	25.9808	8.7721
676	45 69 76	308 915 776	26.0000	8.7764
677	45 83 29	310 288 733	26.0192	8.7807
678	45 96 84	311 665 752	26.0384	8.7850
679	46 10 41	313 046 839	26.0576	8.7893
680	46 24 00	314 432 000	26.0768	8.7937
681	46 37 61	315 821 241	26.0960	8.7980
682	46 51 24	317 214 568	26.1151	8.8023
683	46 64 89	318 611 987	26.1343	8.8066
684	46 78 56	320 013 504	26.1534	8.8109
685	46 92 25	321 419 125	26.1725	8.8152
686	47 05 96	322 828 856	26.1916	8.8194
687	47 19 69	324 242 703	26.2107	8.8237
688	47 33 44	325 660 672	26.2298	8.8280
689	47 47 21	327 082 769	26.2488	8.8323
690	47 61 00	328 509 000	26.2679	8.8366
691	47 74 81	329 939 371	26.2869	8.8408
692	47 88 64	331 373 888	26.3059	8.8451
693	48 02 49	332 812 557	26.3249	8.8493
694	48 16 36	334 255 384	26.3439	8.8536
695	48 30 25	335 702 375	26.3629	8.8578
696	48 44 16	337 153 536	26.3818	8.8621
697	48 58 09	338 608 873	26.4008	8.8663
698	48 72 04	340 068 392	26.4197	8.8706
699	48 86 01	341 532 099	26.4386	8.8748
700	49 00 00	343 000 000	26.4575	8.8790

Number	Square.	Cube.	Square Root.	Cube Root.
701	49 14 01	344 472 101	26.4764	8.8833
702	49 28 04	345 948 408	26.4953	8.8875
703	49 42 09	347 428 927	26.5141	8.8917
704	49 56 16	348 913 664	26.5330	8.8959
705	49 70 25	350 402 625	26.5518	8.9001
706	49 84 36	351 895 816	26.5707	8.9043
707	49 98 49	353 393 243	26.5895	8.9085
708	50 12 64	354 894 912	26.6083	8.9127
709	50 26 81	356 400 829	26.6271	8.9169
710	50 41 00	357 911 000	26.6458	8.9211
711	50 55 21	359 425 431	26.6646	8.9253
712	50 69 44	360 944 128	26.6833	8.9295
713	50 83 69	362 467 097	26.7021	8.9337
714	50 97 96	363 994 344	26.7208	8.9378
715	51 12 25	365 525 875	26.7395	8.9420
716	51 26 56	367 061 696	26.7582	8.9462
717	51 40 89	368 601 813	26.7769	8.9503
718	51 55 24	370 146 232	26.7955	8.9545
719	51 69 61	371 694 959	26.8142	8.9587
720	51 84 00	373 248 000	26.8328	8.9628
721	51 98 41	374 805 361	26.8514	8.9670
722	52 12 84	376 367 048	26.8701	8.9711
723	52 27 29	377 933 067	26.8887	8.9752
724	52 41 76	379 503 424	26.9072	8.9794
725	52 56 25	381 078 125	26.9258	8.9835
726	52 70 76	382 657 176	26.9444	8.9876
727	52 85 29	384 240 583	26.9629	8.9918
728	52 99 84	385 828 352	26.9815	8.9959
729	53 14 41	387 420 489	27.0000	9.0000
730	53 29 00	389 017 000	27.0185	9.0041
731	53 43 61	390 617 891	27.0370	9.0082
732	53 58 24	392 223 168	27.0555	9.0123
733	53 72 89	393 832 837	27.0740	9.0164
734	53 87 56	395 446 904	27.0924	9.0205
735	54 02 25	397 065 375	27.1109	9.0246
736	54 16 96	398 688 256	27.1293	9.0287
737	54 31 69	400 315 553	27.1477	9.0328
738	54 46 44	401 947 272	27.1662	9.0369
739	54 61 21	403 583 419	27.1846	9.0410
740	54 76 00	405 224 000	27.2029	9.0450
741	54 90 81	406 869 021	27.2213	9.0491
742	55 05 64	408 518 488	27.2397	9.0532
743	55 20 49	410 172 407	27.2580	9.0572
744	55 35 36	411 830 784	27.2764	9.0613
745	55 50 25	413 493 625	27.2947	9.0654
746	55 65 16	415 160 936	27.3130	9.0694
747	55 80 09	416 832 723	27.3313	9.0735
748	55 95 04	418 508 992	27.3496	9.0775
749	56 10 01	420 189 749	27.3679	9.0816
750	56 25 00	421 875 000	27.3861	9.0856

Number	Square.	Cube.	Square Root.	Cube Root.
751	56 40 01	423 564 751	27.4044	9.0896
752	56 55 04	425 259 008	27.4226	9.0937
753	56 70 09	426 957 777	27.4408	9.0977
754	56 85 16	428 661 064	27.4591	9.1017
755	57 00 25	430 368 875	27.4773	9.1057
756	57 15 36	432 081 216	27.4955	9.1098
757	57 30 49	433 798 093	27.5136	9.1138
758	57 45 64	435 519 512	27.5318	9.1178
759	57 60 81	437 245 479	27.5500	9.1218
760	57 76 00	438 976 000	27.5681	9.1258
761	57 91 21	440 711 081	27.5862	9.1298
762	58 06 44	442 450 728	27.6043	9.1338
763	58 21 69	444 194 947	27.6225	9.1378
764	58 36 96	445 943 744	27.6405	9.1418
765	58 52 25	447 697 125	27.6586	9.1458
766	58 67 56	449 455 096	27.6767	9.1498
767	58 82 89	451 217 663	27.6948	9.1537
768	58 98 24	452 984 832	27.7128	9.1577
769	59 13 61	454 756 609	27.7308	9.1617
770	59 29 00	456 533 000	27.7489	9.1657
771	59 44 41	458 314 011	27.7669	9.1696
772	59 59 84	460 099 648	27.7849	9.1736
773	59 75 29	461 889 917	27.8029	9.1775
774	59 90 76	463 684 824	27.8209	9.1815
775	60 06 25	465 484 375	27.8388	9.1855
776	60 21 76	467 288 576	27.8568	9.1894
777	60 37 29	469 097 433	27.8747	9.1933
778	60 52 84	470 910 952	27.8927	9.1973
779	60 68 41	472 729 139	27.9106	9.2012
780	60 84 00	474 552 000	27.9285	9.2052
781	60 99 61	476 379 541	27.9464	9.2091
782	61 15 24	478 211 768	27.9643	9.2130
783	61 30 89	480 048 687	27.9821	9.2170
784	61 46 56	481 890 304	28.0000	9.2209
785	61 62 25	483 736 625	28.0179	9.2248
786	61 77 96	485 587 656	28.0357	9.2287
787	61 93 69	487 443 403	28.0535	9.2326
788	62 09 44	489 303 872	28.0713	9.2365
789	62 25 21	491 169 069	28.0891	9.2404
790	62 41 00	493 039 000	28.1069	9.2443
791	62 56 81	494 913 671	28.1247	9.2482
792	62 72 64	496 793 088	28.1425	9.2521
793	62 88 49	498 677 257	28.1603	9.2560
794	63 04 36	500 566 184	28.1780	9.2599
795	63 20 25	502 459 875	28.1957	9.2638
796	63 36 16	504 358 336	28.2135	9.2677
797	63 52 09	506 261 573	28.2312	9.2716
798	63 68 04	508 169 592	28.2489	9.2754
799	63 84 01	510 082 399	28.2666	9.2793
800	64 00 00	512 000 000	28.2843	9.2832

Numbe·	Square.	Cube.	Square Root.	Cube Root
801	64 16 01	513 922 401	28.3019	9.2870
802	64 32 04	515 849 608	28.3196	9.2909
803	64 48 09	517 781 627	28.3373	9.2948
804	64 64 16	519 718 464	28.3549	9.2986
805	64 80 25	521 660 125	28.3725	9.3025
806	64 96 36	523 606 616	28.3901	9.3063
807	65 12 49	525 557 943	28.4077	9.3102
808	65 28 64	527 514 112	28.4253	9.3140
809	65 44 81	529 475 129	28.4429	9.3179
810	65 61 00	531 441 000	28.4605	9.3217
811	65 77 21	533 411 731	28.4781	9.3255
812	65 93 44	535 387 328	28.4956	9.3294
813	66 09 69	537 367 797	28.5132	9.3332
814	66 25 96	539 353 144	28.5307	9.3370
815	66 42 25	541 343 375	28.5482	9.3408
816	66 58 56	543 338 496	28.5657	9.3447
817	66 74 89	545 338 513	28.5832	9.3485
818	66 91 24	547 343 432	28.6007	9.3523
819	67 07 61	549 353 259	28.6182	9.3561
820	67 24 00	551 368 000	28.6356	9.3599
821	67 40 41	553 387 661	28 6531	9.3637
822	67 56 84	555 412 248	28.6705	9.3675
823	67 73 29	557 441 767	28.6880	9.3713
824	67 89 76	559 476 224	28 7054	9.3751
825	68 06 25	561 515 625	28.7228	9.3789
826	68 22 76	563 559 976	28.7402	9.3827
827	68 39 29	565 609 283	28.7576	9.3865
828	68 55 84	567 663 552	28.7750	9.3902
829	68 72 41	569 722 789	28.7924	9.3940
830	68 89 00	571 787 000	28.8097	9.3978
831	69 05 61	573 856 191	28.8271	9.4016
832	69 22 24	575 930 368	28.8444	9.4053
833	69 38 89	578 009 537	28.8617	9.4091
834	69 55 56	580 093 704	28.8791	9.4129
835	69 72 25	582 182 875	28.8964	9.4166
836	69 88 96	584 277 056	28.9137	9.4204
837	70 05 69	586 376 253	28.9310	9.4241
838	70 22 44	588 480 472	28.9482	9.4279
839	70 39 21	590 589 719	28.9655	9.4316
840	70 56 00	592 704 000	28.9828	9.4354
841	70 72 81	594 823 321	29.0000	9.4391
842	70 89 64	596 947 688	29.0172	9.4429
843	71 06 49	599 077 107	29.0345	9.4466
844	71 23 36	601 211 584	29.0517	9.4503
845	71 40 25	603 351 125	29.0689	9.4541
846	71 57 16	605 495 736	29.0861	9.4578
847	71 74 09	607 645 423	29.1033	9.4615
848	71 91 04	609 800 192	29.1204	9.4652
849	72 08 01	611 960 049	29.1376	9.4690
850	72 25 00	614 125 000	29.1548	9.4727

Number	Square.	Cube.	Square Root	Cube Root.
851	72 42 01	616 295 051	29.1719	9.4764
852	72 59 04	618 470 208	29.1890	9.4801
853	72 76 09	620 650 477	29.2062	9.4838
854	72 93 16	622 835 864	29.2233	9.4875
855	73 10 25	625 026 375	29.2404	9.4912
856	73 27 36	627 222 016	29.2575	9.4949
857	73 44 49	629 422 793	29.2746	9.4986
858	73 61 64	631 628 712	29.2916	9.5023
859	73 78 81	633 839 779	29.3087	9.5060
860	73 96 00	636 056 000	29.3258	9.5097
861	74 13 21	638 277 381	29.3428	9.5134
862	74 30 44	640 503 928	29.3598	9.5171
863	74 47 69	642 735 647	29.3769	9.5207
864	74 64 96	644 972 544	29.3939	9.5244
865	74 82 25	647 214 625	29.4109	9.5281
866	74 99 56	649 461 896	29.4279	9.5317
867	75 16 89	651 714 363	29.4449	9.5354
868	75 34 24	653 972 032	29.4618	9.5391
869	75 51 61	656 234 909	29.4788	9.5427
870	75 69 00	658 503 000	29.4958	9.5464
871	75 86 41	660 776 311	29.5127	9.5501
872	76 03 84	663 054 848	29.5296	9.5537
873	76 21 29	665 338 617	29.5466	9.5574
874	76 38 76	667 627 624	29.5635	9.5610
875	76 56 25	669 921 875	29.5804	9.5647
876	76 73 76	672 221 376	29.5973	9.5683
877	76 91 29	674 526 133	29.6142	9.5719
878	77 08 84	676 836 152	29.6311	9.5756
879	77 26 41	679 151 439	29.6479	9.5792
880	77 44 00	681 472 000	29.6648	9.5828
881	77 61 61	683 797 841	29.6816	9.5865
882	77 79 24	686 128 968	29.6985	9.5901
883	77 96 89	688 465 387	29.7153	9.5937
884	78 14 56	690 807 104	29.7321	9.5973
885	78 32 25	693 154 125	29.7489	9.6010
886	78 49 96	695 506 456	29.7658	9.6046
887	78 67 69	697 864 103	29.7825	9.6082
888	78 85 44	700 227 072	29.7993	9.6118
889	79 03 21	702 595 369	29.8161	9.6154
890	79 21 00	704 969 000	29.8329	9.6190
891	79 38 81	707 347 971	29.8496	9.6226
892	79 56 64	709 732 288	29.8664	9.6262
893	79 74 49	712 121 957	29.8831	9.6298
894	79 92 36	714 516 984	29.8998	9.6334
895	80 10 25	716 917 375	29.9166	9.6370
896	80 28 16	719 323 136	29.9333	9.6406
897	80 46 09	721 734 273	29.9500	9.6442
898	80 64 04	724 150 792	29.9666	9.6477
899	80 82 01	726 572 699	29.9833	9.6513
900	81 00 00	729 000 000	30.0000	9.6549

Number	Square.	Cube.	Square Root.	Cube Root.
901	81 18 01	731 432 701	30.0167	9.6585
902	81 36 04	733 870 808	30.0333	9.6620
903	81 54 09	736 314 327	30.0500	9.6656
904	81 72 16	738 763 264	30.0666	9.6692
905	81 90 25	741 217 625	30.0832	9.6727
906	82 08 36	743 677 416	30.0998	9.6763
907	82 26 49	746 142 643	30.1164	9.6799
908	82 44 64	748 613 312	30.1330	9.6834
909	82 62 81	751 089 429	30.1496	9.6870
910	82 81 00	753 571 000	30.1662	9.6905
911	82 99 21	756 058 031	30.1828	9.6941
912	83 17 44	758 550 825	30.1993	9.6976
913	83 35 69	761 048 497	30.2159	9.7012
914	83 53 96	763 551 944	30.2324	9.7047
915	83 72 25	766 060 875	30.2490	9.7082
916	83 90 56	768 575 296	30.2655	9.7118
917	84 08 89	771 095 213	30.2820	9.7153
918	84 27 24	773 620 632	30.2985	9.7188
919	84 45 61	776 151 559	30.3150	9.7224
920	84 64 00	778 688 000	30.3315	9.7259
921	84 82 41	781 229 961	30.3480	9.7294
922	85 00 84	783 777 448	30.3645	9.7329
923	85 19 29	786 330 467	30.3809	9.7364
924	85 37 76	788 889 024	30.3974	9.7400
925	85 56 25	791 453 125	30.4138	9.7435
926	85 74 76	794 022 776	30.4302	9.7470
927	85 93 29	796 597 983	30.4467	9.7505
928	86 11 84	799 178 752	30.4631	9.7540
929	86 30 41	801 765 089	30.4795	9.7575
930	86 49 00	804 357 000	30.4959	9.7610
931	86 67 61	806 954 491	30.5123	9.7645
932	86 86 24	809 557 568	30.5287	9.7680
933	87 04 89	812 166 237	30.5450	9.7715
934	87 23 56	814 780 504	30.5614	9.7750
935	87 42 25	817 400 375	30.5778	9.7785
936	87 60 96	820 025 856	30.5941	9.7819
937	87 79 69	822 656 953	30.6105	9.7854
938	87 98 44	825 293 672	30.6268	9.7889
939	88 17 21	827 936 019	30.6431	9.7924
940	88 36 00	830 584 000	30.6594	9.7959
941	88 54 81	833 237 621	30.6757	9.7993
942	88 73 64	835 896 888	30.6920	9.8028
943	88 92 49	838 561 807	30.7083	9.8063
944	89 11 36	841 232 384	30.7246	9.8097
945	89 30 25	843 908 625	30.7409	9.8132
946	89 49 16	846 590 536	30.7571	9.8167
947	89 68 09	849 278 123	30.7734	9.8201
948	89 87 04	851 971 392	30.7896	9.8236
949	90 06 01	854 670 349	30.8058	9.8270
950	90 25 00	857 375 000	30.8221	9.8305

Number	Square.	Cube.	Square Root.	Cube Root.
951	90 44 01	860 085 351	30.8383	9.8339
952	90 63 04	862 801 408	30.8545	9.8374
953	90 82 09	865 523 177	30.8,07	9.8408
954	91 01 16	868 250 664	30.8869	9.8443
955	91 20 25	870 983 875	30.9031	9.8477
956	91 39 36	873 722 816	30.9192	9.8511
957	91 58 49	876 467 493	30.9354	9.8546
958	91 77 64	879 217 912	30.9516	9.8580
959	91 96 81	881 974 079	30.9677	9.8614
960	92 16 00	884 736 000	30.9839	9.8648
961	92 3 21	887 503 681	31.0000	9.8683
962	92 54 44	890 277 128	31.0161	9.8717
963	92 73 69	893 056 347	31.0322	9.8751
964	92 92 96	895 841 344	31.0483	9.8785
965	93 12 25	898 632 125	31.0644	9.8819
966	93 31 56	901 428 696	31.0805	9.8854
967	93 50 89	904 231 063	31.0966	9.8888
968	93 70 24	907 039 232	31.1127	9.8922
969	93 89 61	909 853 209	31.1288	9.8956
970	94 09 00	912 673 000	31.1448	9.8990
971	94 28 41	915 498 6.1	31.1609	9.9024
972	94 47 84	918 330 048	31.1769	9.9058
973	94 67 29	921 167 317	31.1929	9.9092
974	94 86 76	924 010 424	31.2090	9.9126
975	95 06 25	926 859 375	31.2250	9.9160
976	95 25 76	929 714 176	31.2410	9.9194
977	95 45 29	932 574 833	31.2570	9.9227
978	95 64 84	9?5 441 352	31.2730	9.9261
979	95 84 41	938 313 739	31.2890	9.9295
980	96 04 00	941 192 000	31.3050	9.9329
981	96 23 61	944 076 141	31.3209	9.9363
982	96 43 24	946 966 168	31.3369	9.9396
983	96 62 89	949 862 087	31.3528	9.9430
984	96 82 56	952 763 904	31.3688	9.9464
985	97 02 25	9?5 671 625	31.3847	9.9497
986	97 21 96	958 585 256	31.4006	9.9531
987	97 41 69	961 504 803	31.4166	9.9565
988	97 61 44	964 430 272	31.4325	9.9598
989	97 81 21	967 361 669	31.4484	9.9632
990	98 01 00	970 299 000	31.4643	9.9666
991	98 20 81	973 242 271	31.4802	9.9699
992	98 40 64	976 191 488	31.4960	9.9733
993	98 60 49	979 146 657	31.5119	9.9766
994	98 80 36	982 107 784	31.5278	9.9800
995	99 00 25	985 074 875	31.5436	9.9833
996	99 20 16	988 047 936	31.5595	9.9866
997	99 40 09	991 026 973	31.5753	9.9900
998	99 60 04	994 011 992	31.5911	9.9933
999	99 80 01	997 002 999	31.6070	9.9967
1000	1 00 00 00	1 000 000 000	31.6228	10.0000

WIRE AND SHEET-METAL GAUGES COMPARED.

Number of Gauge.	Birmingham (or Stubs' Iron) Wire Gauge.	American or Brown and Sharpe Gauge.	Roebling's and Washburn & Moen's Gauge.	Stubs' Steel Wire Gauge. (See also p. 29.)	British Imperial Standard Wire Gauge. (Legal Standard in Great Britain since March 1, 1884.)		U. S. Standard Gauge for Sheet and Plate Iron and Steel. (Legal Standard since July 1, 1893.)	Number of Gauge.
	inch.	inch.	inch.	inch.	inch.	millim.	inch.	
0000000			.49		.500	12.7	.5	7/0
000000			.46		.464	11.78	.469	6/0
00000			.43		.432	10.97	.438	5/0
0000	.454	.46	.393		.4	10.16	.406	4/0
000	.425	.40964	.362		.372	9.45	.375	3/0
00	.38	.3648	.331		.348	8.84	.344	2/0
0	.34	.32486	.307		.324	8.23	.313	0
1	.3	.2893	.283	.227	.3	7.62	.281	1
2	.284	.25763	.263	.219	.276	7.01	.266	2
3	.259	.22942	.244	.212	.252	6.4	.25	3
4	.238	.20431	.225	.207	.232	5.89	.234	4
5	.22	.18194	.207	.204	.212	5.38	.219	5
6	.203	.16202	.192	.201	.192	4.88	.203	6
7	.18	.14428	.177	.199	.176	4.47	.188	7
8	.165	.12849	.162	.197	.16	4.06	.172	8
9	.148	.11443	.148	.194	.144	3.66	.156	9
10	.134	.10189	.135	.191	.128	3.25	.141	10
11	.12	.09074	.12	.188	.116	2.95	.125	11
12	.109	.08081	.105	.185	.104	2.64	.109	12
13	.095	.07196	.092	.182	.092	2.34	.094	13
14	.083	.06408	.08	.180	.08	2.03	.078	14
15	.072	.05707	.072	.178	.072	1.83	.07	15
16	.065	.05082	.063	.175	.064	1.63	.0625	16
17	.058	.04526	.054	.172	.056	1.42	.0563	17
18	.049	.0403	.047	.168	.048	1.22	.05	18
19	.042	.03589	.041	.164	.04	1.02	.0438	19
20	.035	.03196	.035	.161	.036	.91	.0375	20
21	.032	.02846	.032	.157	.032	.81	.0344	21
22	.028	.02535	.028	.155	.028	.71	.0313	22
23	.025	.02257	.025	.153	.024	.61	.0281	23
24	.022	.0201	.023	.151	.022	.56	.025	24
25	.02	.0179	.02	.148	.02	.51	.0219	25
26	.018	.01594	.018	.146	.018	.46	.0188	26
27	.016	.01419	.017	.143	.0164	.42	.0172	27
28	.014	.01264	.016	.139	.0148	.38	.0156	28
29	.013	.01126	.015	.134	.0136	.35	.0141	29
30	.012	.01002	.014	.127	.0124	.31	.0125	30
31	.01	.00893	.0135	.120	.0116	.29	.0109	31
32	.009	.00795	.013	.115	.0108	.27	.0101	32
33	.008	.00708	.011	.112	.01	.25	.0094	33
34	.007	.0063	.01	.110	.0092	.23	.0086	34
35	.005	.00561	.0095	.108	.0084	.21	.0078	35
36	.004	.005	.009	.106	.0076	.19	.007	36
37		.00445	.0085	.103	.0068	.17	.0066	37
38		.00396	.008	.101	.006	.15	.0063	38
39		.00353	.0075	.099	.0052	.13		39
40		.00314	.007	.097	.0048	.12		40
41				.095	.0044	.11		41
42				.092	.004	.10		42
43				.088	.0036	.09		43
44				.085	.0032	.08		44
45				.081	.0028	.07		45
46				.079	.0024	.06		46
47				.077	.002	.05		47
48				.075	.0016	.04		48
49				.072	.0012	.03		49
50				.069	.001	.025		50

CHAPTER I.*

SCREWS, NUTS, AND BOLTS.

A Screw is a helical projection or thread formed upon a cylinder and is the most common device used in mechanical

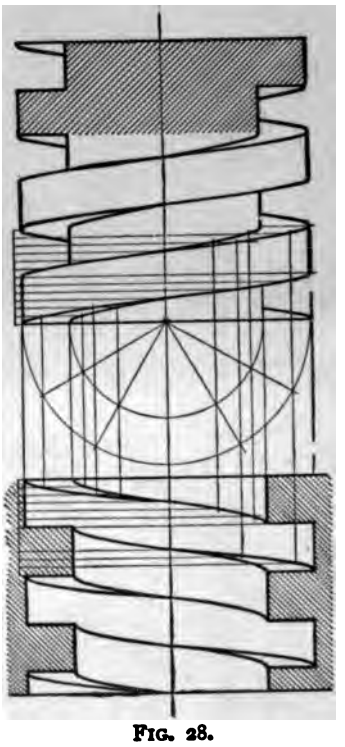


FIG. 28.

combinations. It is employed in the construction of machinery for producing pressure contact and transmitting

* Copyright.

motion. When the thread of an external screw is made to fit into the corresponding hollow of an internal screw (Fig. 28) the latter is termed its nut.

The Pitch of a Screw-thread is the lineal distance its nut would advance along the axis in one turn. In a single-threaded screw the pitch is the distance between the centres of two consecutive threads measured in the direction of the axis, in a double-threaded screw it is the distance from centre to centre of every alternate thread, and in a triple-threaded screw it is a distance that will embrace three threads. For screw-fastenings, instead of giving the pitch the number of threads per inch of screw is given—for example, a bolt of 1″ diameter has generally 8 threads per inch; this means that the bolt has a single thread wound around it 8 times for every inch of its length.

Right- and Left-handed Screws.—Screws are made right- and left-handed, of which the right-handed are the more common and are distinguished by their nuts advancing along the screws when turned in the direction in which the hands of a watch revolve. On a drawing the right-handed screws are distinguished by the threads inclining upwards towards the right hand when the screws are in a vertical position, as in Fig. 28. When a nut with a right-handed thread is shown in section the direction of the threads in the nut is the opposite to the threads on the screw.

The Nominal Diameter of a Screw is the diameter over the tops of the threads and is equal to the diameter of the cylinder upon which the thread is cut. It is the area of the nominal diameter that is considered when estimating the shearing strength.

The Effective Diameter is the diameter at the bottom of the thread and is equal to the diameter of the hole in the nut before its threads are cut. Unless when the bolts are subjected to a shearing stress, it is the area of the effective diameter that is considered in estimating their strength.

The Depth of the Thread is the distance measured perpendicularly to the axis of the screw from the top to the bottom of the thread.

NOTATION.

d = nominal diameter of bolt;
d_1 = effective diameter of bolt;
δ = depth of thread;
δ_1 = total depth of V;
p = pitch of thread;
n = number of threads per inch.

The Forms of Screw-threads in general use in machine construction are represented in Figs. 29–33. The V thread is adopted on all screw-fastenings because of the shearing strength of the threads and frictional holding power, which is due to the normal pressure on the thread being inclined

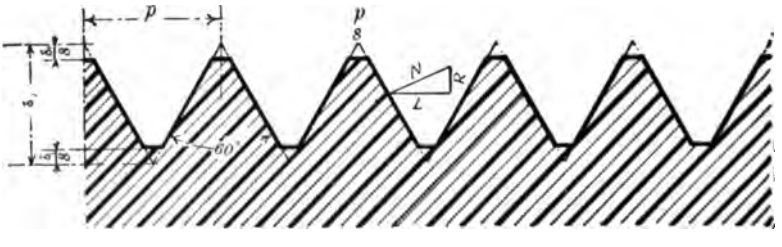


FIG. 29

to the axis of the screw. This normal force N, Fig. 29, may be resolved into two components, one L parallel to the

axis of the screw, and the other R at right angles to it. L represents the load carried by the thread and R the force tending to burst the nut; therefore the greater the angle of the V the greater will be the normal component or bursting force, and, the friction being proportional to the normal force, it will increase with the angle of the V. Of the forms of V threads shown two (Figs. 29 and 33) are in common use in the United States for bolts and nuts.

The Sellers or United States Standard, a section of which is shown in Fig. 29, has been adopted by the U. S. Government, the Railway Master Mechanics' Association, the Master Car-builders' Association, and many of the principal manufactories in this country. The sides of this thread form an angle of 60° with each other, and are $\frac{1}{8}$ of δ_1 short of meeting at a sharp point at the tops and bottoms, which makes the sides of the thread in length equal to $\frac{3}{4}$ of the pitch, and the depth of thread δ will be expressed by the formula

$$\delta = \tfrac{3}{4} \times p \sin 60° = 0.65p. \quad . \quad . \quad . \quad . \quad (1)$$

The effective diameter will then be

$$d_1 = d - 2\delta = d - 1.3p = d - \frac{1.3}{n}. \quad . \quad . \quad (2)$$

The relation between the pitch and the diameter will be expressed by the formula

$$p = 0.24\sqrt{d + 0.625} - 0.175. \quad . \quad . \quad . \quad (3)$$

The number of threads per inch is

$$n = \frac{1}{p} = \frac{1}{0.24\sqrt{d + 0.625} - 0.175}. \quad . \quad . \quad (4)$$

The table of proportions on page 70 has been deduced from the preceding formulæ. A difference, however, may be found between the formulæ and the table in the number of threads per inch, as the table has been modified to avoid as far as practicable troublesome combinations in the gears of screw-cutting machines.

Exercise 1.—Draw 6 threads in sectional outline, of the *Sellers thread* (Fig. 29), suitable for a screw 6″ in diameter. *Scale three times full size.*

Construction.—Begin by drawing a horizontal line in the upper left-hand corner of the paper $\frac{3}{4}$″ down from the border-line, and a vertical line about $\frac{3}{4}$″ in from the left-hand border-line. Then find the pitch p by the formula (3), and from where the two lines you have just drawn intersect mark off with the scale on the horizontal line 6 points a distance apart equal to the pitch as found by the formula. Through these points with the 30° triangle draw the V's. Complete the pencilling by dividing the depth of the V into 8 equal divisions, and cut off one division at the top and bottom of each thread.

The Sharp V Thread, shown in Fig. 30, is one of the

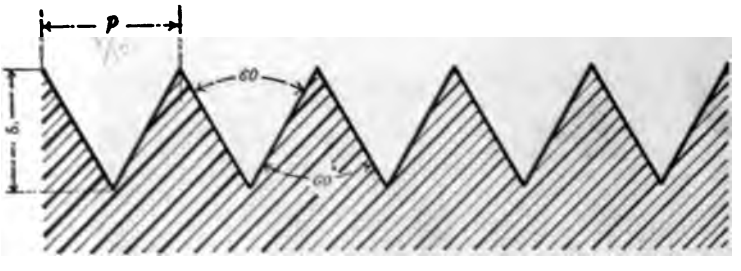


FIG. 30.

forms of threads that were in use before the Sellers thread

was adopted as the U. S. standard, and is still used, although condemned by all progressive engineers. This thread is the same as the Sellers thread except that the sides are made to meet at a sharp point at the top and bottom, which makes the sides of the thread equal in length to the pitch p, and the depth of the thread δ_1 will be expressed by the formula

$$\delta_1 = p \sin 60° = 0.866p. \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The effective diameter of the bolt (d_1) will then be expressed by the formula

$$d_1 = d - 2 \times 0.866p = d - 1.732. \quad . \quad . \quad (6)$$

Now, comparing the effective diameters, we have:

U. S. threads	$d_1 = d - 1.3p.$	(2)
V threads	$d_1 = d - 1.732p.$	(6)

This serves to show that with an equal pitch the *effective diameter* of the screw having a U. S. standard thread is greater than one with a sharp V thread. While the latter form of thread materially diminishes the strength of the bolt, the sharp point adds very little strength to the thread. A further objection to this form of thread is the variation in depth of the threads due to the wear of the sharp points on the taps and dies used in producing them.

The Whitworth V Thread, an outline section of which is shown in Fig. 31, is the British standard, and is generally adopted on all screw-fastenings in British machine construction. It has the sides of the V inclined to each other at an angle of 55°, and has an amount rounded off at the top and bottom equal to $\frac{1}{6}$ of the total depth of the V. *The table of*

dimensions for Whitworth screws (page 70) has been deduced from the following formulæ. The total depth of the V

$$\delta_1 = 0.5 \cot 27\tfrac{1}{2}^\circ = 0.96p. \quad . \quad . \quad . \quad . \quad (7)$$

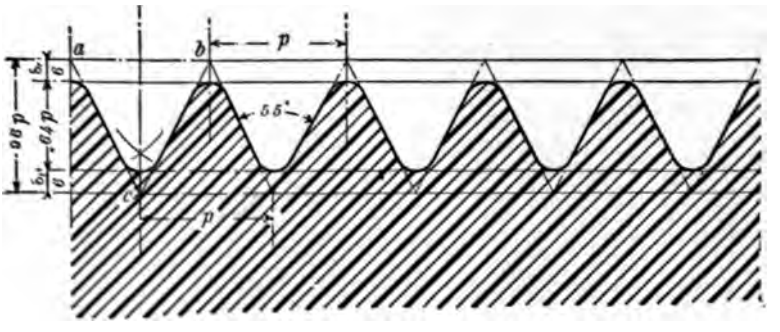


FIG. 31.

The depth of the finished thread

$$\delta = \tfrac{8}{8} \times 0.96p = 0.64p. \quad . \quad . \quad . \quad . \quad (8)$$

The pitch $p = 0.08d + 0.04$. (9)

Number of threads per inch

$$= \frac{1}{p} \quad \text{and} \quad p = \frac{1}{n}. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The diameter at the bottom of the thread will be given by the formula

$$d_1 = d - 2 \times 0.64p = d - \frac{1.28}{n}. \quad . \quad . \quad . \quad (11)$$

Exercise 2.—Draw 6 threads of the *Whitworth form of thread* (Fig. 31). Pitch $\frac{1}{2}''$. *Scale three times full size.*

Construction.—At a suitable distance below the drawing of the Sellers thread draw two horizontal lines parallel to each other and a distance apart equal to $0.96p$. On the lower line mark off a distance *ab* equal to the pitch. Bisect

ab and draw the bisecting line to cut the lower parallel line at the point *c*. Join *ca* and *cb*, which will be inclined to each other at an angle of 55°. Mark off the pitch from *b* along the upper line, and from *c* along the lower line, to give the required number of threads. Complete the pencilling by rounding off the sharp points of the V.

The Square Screw-thread.—The form of thread which is invariably called the square thread is really a rectangle, the depth of the thread being equal to $0.485p$ and its width equal to $0.5p$. However, it is usual and accurate enough to make it square upon the drawing. *On screws of the same diameter the pitch of a square-threaded screw is usually made equal to twice the pitch of one with a V thread; therefore the square thread will have only half the amount of material at the bottom of the thread that the V thread has to resist the shearing action of the load. As the bearing-surfaces of this screw are perpendicular to the axis, and the force applied parallel to it, there will be no bursting force upon the nut; and as the reaction is nearly equal to the load on the square-threaded screw, there will be less friction than there is under the same conditions with a V thread; consequently the square thread is best adapted for transmitting motion when the load has to be moved in opposite directions.

The Knuckle or Rounded Screw-thread is a modification of the square thread in which the top and bottom of each thread are made semicircular, as shown in Fig. 32. This form of thread is used for rough work and can be readily thrown in and out of gear with a portion of a nut.

The Buttress Screw-thread is a combination of the V and square threads, one side being perpendicular, and the

* Klein gives $p = .08 + .09d$, $d_1 = .91d - .08$.

other inclined at an angle of 45° to the axis of the screw, and has an amount cut from the top and bottom of each

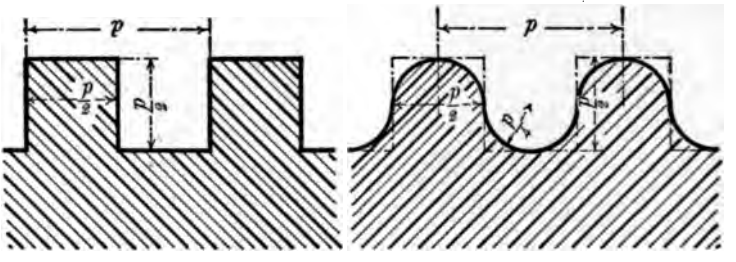


FIG. 32.

thread equal to $\frac{1}{8}$ of the total depth of the thread, as shown in Fig. 33. This form of thread can be used only when the pressure is on that side of the thread which is at right angles to the axis of the screw.

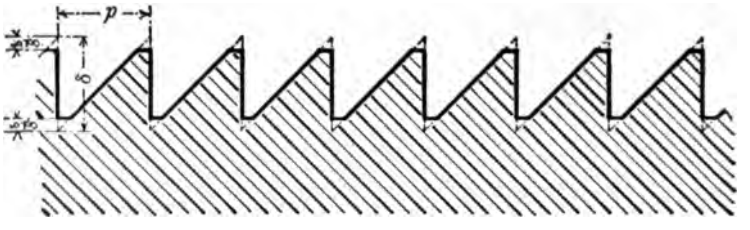


FIG. 33.

Exercise 3.—Draw the sectional outline of the square, knuckle, and buttress threads shown in Figs. 32 and 33. Pitch 1″. *Scale twice full size.*

Pipe-threads.—Previous to the year 1862 no common system had been agreed upon for the form or proportions of pipe-threads. Since that time, owing to the efforts of the late Robert Briggs, C.E., who proposed formulæ and tables for the dimensions of pipes and pipe-threads, a standard

TABLE 1.

STANDARD DIMENSIONS OF WROUGHT-IRON WELDED TUBES.

(BRIGGS STANDARD.)

Diameter of Tube.			Thickness of Metal.	Screwed Ends.	
Nominal Inside.	Actual Inside.	Actual Outside.		Number of Threads per Inch.	Length of Perfect Screw.
Inches	Inches.	Inches.	Inch.	No.	Inches.
⅛	0.270	0.405	0.068	27	0.19
¼	0.364	0.540	0.088	18	0.29
⅜	0.494	0.675	0.091	18	0.30
½	0.623	0.840	0.109	14	0.39
¾	0.824	1.050	0.113	14	0.40
1	1.048	1.315	0.134	11½	0.51
1¼	1.380	1.660	0.140	11	0.54
1½	1.610	1.900	0.145	11½	0.55
2	2.067	2.375	0.154	11½	0.58
2½	2.468	2.875	0.204	8	0.89
3	3.067	3.500	0.217	8	0.95
3½	3.548	4.000	0.226	8	1.00
4	4.026	4.500	0.237	8	1.05
4½	4.508	5.000	0.246	8	1.10
5	5.045	5.563	0.259	8	1.16
6	6.065	6.625	0.280	8	1.26
7	7.023	7.625	0.301	8	1.36
8	7.982	8.625	0.322	8	1.46
9	9.000	9.625	0.344	8	1.57
10	10.019	10.750	0.366	8	1.68

Taper of conical tube-ends, 1 in 32 to axis of tube (¾ in. per foot total taper).

system has been generally used and was formally adopted by the manufacturers of wrought-iron pipes and boiler-tubes and by the Association of Manufacturers of Brass and Iron Steam-, Gas-, and Water-work of the United States.

The following is an extract from a paper by Mr. Briggs as given in the report of the American Society of Engineers:

"The thread employed has an angle of 60°; it is slightly rounded off, both at the top and at the bottom, so that the height or depth of the thread, instead of being exactly equal to the pitch, is only four fifths of the pitch, or equal to $0.8\frac{1}{n}$,

if n be the number of threads per inch. For the length of tube-end throughout which the screw-thread continues perfect the empirical formula used is $T = (0.8D + 4.8) \times \frac{1}{n}$, where D is the actual external diameter of the tube throughout its parallel length, and is expressed in inches. Further back, beyond the perfect threads, come two having the same taper at the bottom, but imperfect at the top. The remaining imperfect portion of the screw-thread, furthest back from the extremity of the tube, is not essential in any way to this system of joint; and its imperfection is simply incidental to the process of cutting the thread at a single operation.

Exercise 4.—Draw a section of a pipe-screw (Fig. 34) for a wrought-iron pipe 8″ in diameter. *Scale five times full size.*

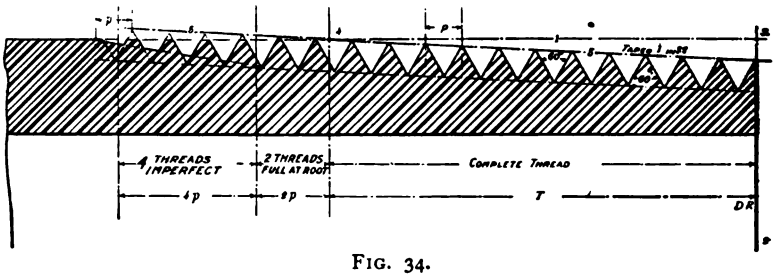


FIG. 34.

Construction.—Draw two lines parallel to each other at a distance apart equal to the thickness of metal as given in the table; then draw the vertical line 2 to represent the end of the pipe, and from 2 along the line 1 mark off 3, 4, equal to T. *Taper 1 in 32 means an inclination of 1 unit in height to every 32 units in length.* From the point 4 draw the line 5 at the required inclination. On the line 5 from where it intersects 2 mark off points at a distance apart equal to the pitch, and through these points with the 30° triangle draw the

threads. The bottoms of the last 4 threads are cut off by drawing a line from the bottom of the last thread that is full at the bottom to a point on the surface of the pipe which is a distance beyond the screwed part equal to the pitch.

Screw-thread Conventions.—The method of drawing screws to represent their true form is shown in Fig. 28, but it is quite obvious that it is unnecessary for the draftsman to perform this lengthy geometrical construction to indicate each screwed piece upon the drawing. Instead he adopts some convention suitable to the class of drawing he is making that can be quickly drawn and is generally understood to represent a screw-thread. Fig. 35, No. 1,

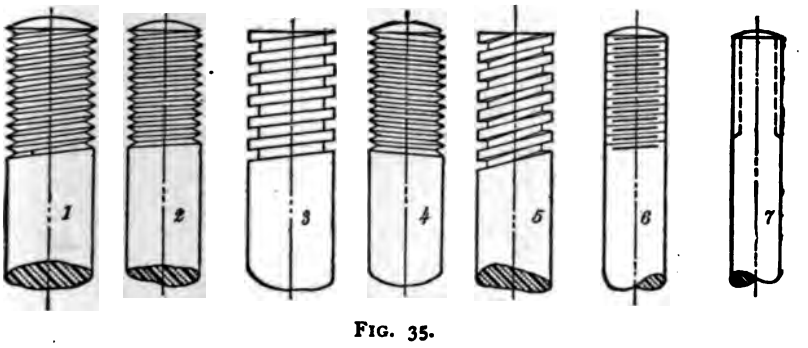


FIG. 35.

shows a convention for a double V thread; No. 2, a single V thread; No. 3, a single square thread; No. 4, a single left-hand V thread; No. 5, a double right-hand square thread; No. 6, any V thread of small diameter; No. 7, any thread of very small diameter. The method adopted on rough drawings and sketches is shown at No. 7. The dotted lines indicate the bottom of the thread, and the distance they extend along the piece the length of the

screwed part. At Nos. 1, 2, 4 are shown conventions adopted upon finished drawings to represent threaded screws of a large diameter and wide pitch. There are various ways of improving the appearance of this convention: one is by shading the lower lines of each thread, as shown in Fig. 37, and another method is to fill in completely the under side of the thread, as shown in Fig. 39. At No. 6 is shown a method adopted on working drawings to represent screw-threads upon pieces of a small diameter or large screws drawn to a small scale. Here the narrow lines indicate the top and the wide lines the bottom of the screw-thread. When a very long screw has to be represented upon a drawing, as is often the case with the square-threaded screw, a few threads are drawn at the beginning of the screwed part, and the length of the screw is indicated by dotted lines drawn from the bottoms of the threads.

The Nut.—The most common application of the screw for producing contact pressure is the bolt, used in conjunction with a nut, of which there are different forms. The form most in use is the hexagonal (Fig. 37).

The standard proportions for hexagonal nuts are:

H = height = diameter of bolt (d).

F = distance across the flats = $1\frac{1}{2}d + \frac{1}{8}$ of an inch.

D = distance across the corners = $(1\frac{1}{2}d + \frac{1}{8}'')$ 1.155.

Fig. 35 shows the true form of the curves when the end of the nut is machined to form a part of a sphere or cone. This rounding or bevelling off of the corners is called chamfering. The radius r of the chamfering is made from $1\frac{1}{2}d$ to $2d$, and the angle a is made from 60° to 45° with the axis of the nut. *When representing nuts upon a drawing they should*

always be drawn to show the distance across the angles, as in Fig. 40.

Exercise 5.—Draw the true curves of a hexagonal nut for a bolt 6″ in diameter when the top of the nut is chamfered

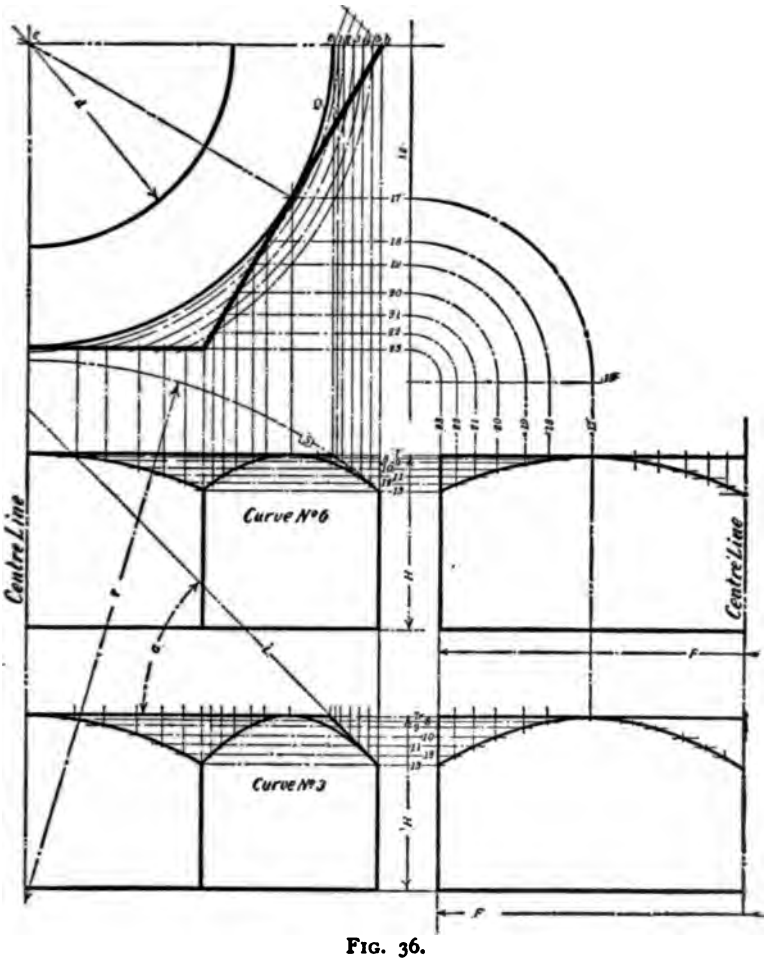


FIG. 36.

off to form a part of a sphere with a radius $r = 1\frac{1}{2}$ times the diameter of the bolt (d), and when the chamfering is a part

of a cone the side of which makes an angle of 45° with the axis of the nut, as shown in Fig. 36.

Construction.—Begin with the plan, first locating the centre *c*, and with *c* as a centre and a radius equal to $\frac{1}{2}d$ draw the quadrant representing the hole in the nut, and from the same centre and a radius equal to half the distance across the flats *F* draw the quadrant *Q*, and on this quadrant circumscribe a part of a hexagon with the 30° triangle and T square, as shown in Fig. 37. Draw the part elevations and end views, and with *r* as a radius and the centre on the centre line draw the arc *S*, which represents the spherical chamfer, and on the lower elevation draw the angle *a*. Divide *eb* into any number of divisions, say 6, at points 1, 2, 3, 4, 5*d*. Where perpendicular lines drawn through these points intersect the arc *S* and line *L* draw the horizontal lines 7, 8, 9, 10, 11, 12, 13, and with *c* as a centre and radii *c*1, *c*2, *c*3, *c*4, *c*5 draw arcs, and from where these arcs intersect the inclined face of the nut draw vertical lines to intersect the lines 7, 8, 9, 10, etc. These points of intersection will be points of the curve on the side face of the nut. The curve of the front face will be an arc of a circle. To find the curves on the side view draw a line 15 say $\frac{1}{2}''$ below and parallel to the lower face of the nut in plan, and a perpendicular line 14 half an inch to the left of the end view; where the arcs drawn through the points 1, 2, 3, etc., from the centre *c* cut the inclined face of the nut in plan draw horizontal lines to intersect the line 14; and with a centre at the intersection of the lines 14 and 15 revolve the lines 17, 18, 19, 20, 21, 22, 23 on to the line 15 and draw perpendicular lines through the points of intersection. The line 17 revolved will be the cen-

tre of the nut face on the end view, and the intersection of the lines 17, 18, 19, 20, 21, 22, 23 with the horizontal lines 7, 8, 9, 10, 11, 12, 13 will be points on one half of the required curve. To complete the curve, with a centre at the intersection of the line 17 and the top of the nut mark with the compasses corresponding points on the other side of the line 17.

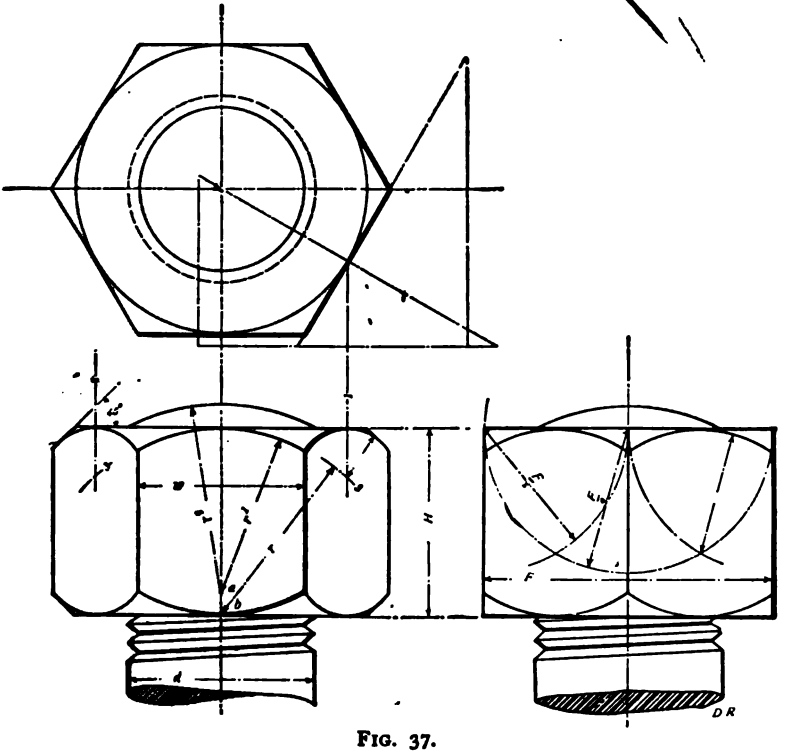


FIG. 37.

A Conventional Method of representing large nuts on drawings is shown in Fig. 37. In this representation the curves of the nut are arcs of circles and the corners are chamfered off at an angle of 45° to the axis of the nut.

Exercise 6.—Draw the three views for a bolt 3″ in diameter. *Scale full size.*

Construction.—Begin, as in the last exercise, by drawing the plan. Locate the centre and draw a circle equal in diameter to the distance across the flats $1\frac{1}{2}d + \frac{1}{8}''$; on this circle with the set-square circumscribe a hexagon, and find the centre of the side faces in the manner shown. Draw the elevation and end view of the hexagon without the curves. With centre *a* and radius r' equal to *w* draw the arc of the middle face tangent to the top of the nut, and with centre *b* and radius *r* equal to *d* draw the arcs 3 to intersect the lines 1 and 2. These points of intersection will be the centres of the arcs of the side face. The method of finding the centres of the curves on the end view is clearly shown on the drawing. Through the points where the outside diameter of the bolt intersects the top of the nut with a radius $r' = d$ draw the arc representing the bolt-point.

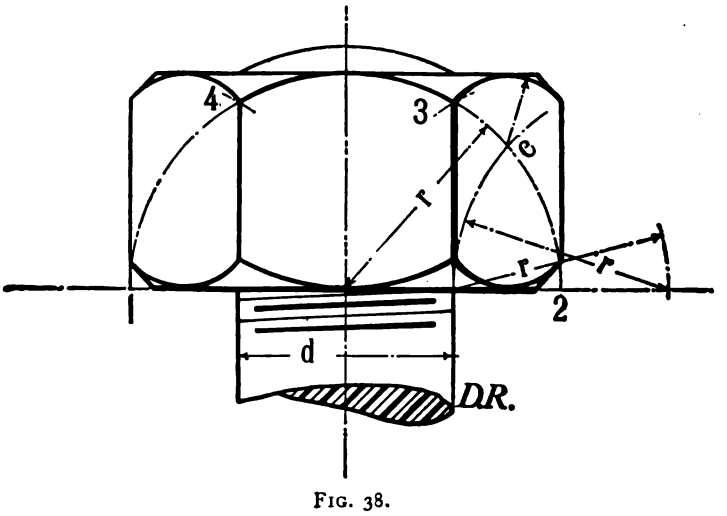


FIG. 38.

When representing small nuts or nuts drawn to a small scale, it is usual to make the distance across the angles $= 2d$. This method does not give the correct proportions and should only be used on nuts and bolt-heads when d is less than 1″ in diameter when drawn to scale. When nuts are chamfered on the upper side only, it is usual to cut the corners parallel to the axis, thus leaving a cylindrical projection on the under side, which bears on the piece the nut is holding, as shown in Fig. 39.

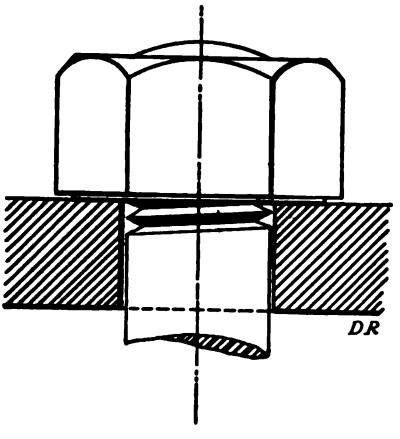


FIG. 39.

The diameter of the cylindrical projection is equal to the distance across the flats ($1\frac{1}{2}d + \frac{1}{8}''$).

Exercise 7.—Draw a hexagonal nut for a bolt $\frac{3}{4}''$ in diameter chamfered on the upper side and finished on the under side as shown in Fig. 39. Make the distance across the angles $= 2d$, and draw the curves by the method shown in Fig. 38. *Scale full size.*

Construction.—Draw the semicircle 1, 2 and divide it into

three equal divisions at the points 1, 2, 3, 4; through these points draw perpendicular lines to intersect the top of the nut. The method of finding the centres of the arcs of the side faces will be clearly understood from Fig. 38, where *r is* in each case $= d$.

Machine fastenings are most commonly effected by means of bolts, keys, or rivets. When two or more pieces have to be held together with the intention of disconnecting them again, a bolt or key is used; the rivet being used only when the connection is to be permanent. The most common form

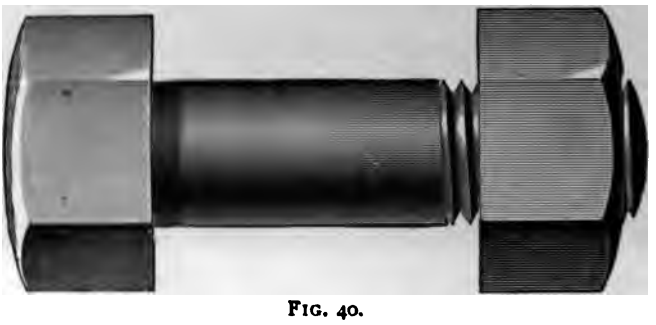


FIG. 40.

of bolt used in general machine construction is the *hexagonal, headed bolt* shown in Fig. 40.

Exercise 8.—Draw a hexagonal headed bolt and nut in position on a cast-iron pipe-flange (Fig. 41). Make the bolt $\frac{3}{4}''$ in diameter. *Scale full size.*

Construction. — First draw the lines representing the thickness of the pipe and flanges. The angles of the nut should be clear of the fillet about $\frac{1}{8}''$ and the radius (r) should be at least $\frac{1}{8}''$. Therefore, the distance (b) will be equal to half the distance across the angles of *the nut* $+ \frac{1}{8}'' + r$. To give the flange a proper finish, the distance (a) is made from

$\frac{1}{8}$" to $\frac{1}{4}$" greater than half the distance across the angles of the nut. The number of threads per inch will be found in Table 8.

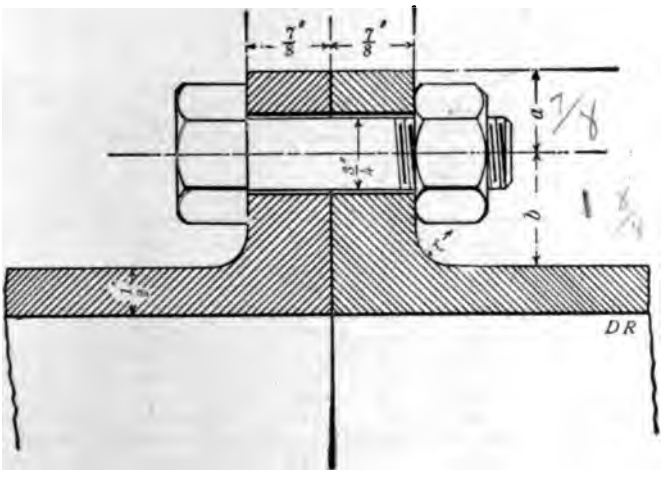


FIG. 41.

The Square-headed Bolt.—Fig. 42 is a cheaper make than the hexagonal and is generally used in structures of rough iron. It is sometimes, however, adopted in machine- and engine-construction generally when the head is let into a recess, as shown in Fig. 42. It is used in this instance in preference to the hexagonal head, because it is easier to make the square recess in the pattern. In Fig. 42, it is shown in combination with a square nut, the sides of which give a better gripping surface for the wrench than the hexagonal, but the latter can be screwed up in a more confined position, as it is only necessary to turn it through an angle of 60° to get the wrench or spanner on to the next two parallel faces; while the square nut has to be turned through an angle of 90° under the same conditions.

TABLE

UNITED STATES STANDARD OF

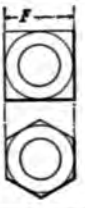
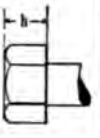
Screw-threads.

Diameter of Screw.	Number of Threads per Inch.	Diameter at Bottom of Threads.	Area at Bottom of Threads in Square Inches.	Area of Bolt Body in Square Inches.
¼	20	.185	.027	.049
5/16	18	.240	.045	.077
⅜	16	.294	.068	.110
7/16	14	.344	.093	.150
½	13	.400	.126	.196
9/16	12	.454	.162	.249
⅝	11	.507	.202	.307
¾	10	.620	.302	.442
⅞	9	.731	.420	.601
1	8	.837	.550	.785
1⅛	7	.940	.694	.994
1¼	7	1.065	.893	1.227
1⅜	6	1.160	1.057	1.485
1½	6	1.284	1.295	1.767
1⅝	5½	1.389	1.515	2.074
1¾	5	1.491	1.746	2.405
1⅞	5	1.616	2.051	2.761
2	4½	1.712	2.302	3.142
2¼	4½	1.962	3.023	3.976
2½	4	2.176	3.719	4.909
2¾	4	2.426	4.620	5.940
3	3½	2.629	5.428	7.069
3¼	3½	2.879	6.510	8.296
3½	3¼	3.100	7.548	9.621
3¾	3	3.317	8.641	11.045
4	3	3.567	9.963	12.566
4¼	2⅞	3.798	11.329	14.186
4½	2¾	4.028	12.753	15.904
4¾	2⅝	4.256	14.226	17.721
5	2½	4.480	15.763	19.635
5¼	2½	4.730	17.572	21.648
5½	2⅜	4.953	19.267	23.758
5¾	2⅜	5.203	21.262	25.967
6	2¼	5.423	23.098	28.274

NOTE.—The above table gives the sizes of the rough nuts and bolt-heads. The finished

8.

SCREW-THREADS, BOLTS, AND NUTS.

Nuts.				Heads.		Tap Drill.
						
1/4	1/2	37/64	7/10	1/4	1/4	3/16
5/16	19/32	11/16	10/12	5/16	19/64	1/4
3/8	11/16	51/64	63/64	3/8	11/32	5/16
7/16	25/32	9/10	1 1/64	7/16	25/64	23/64
1/2	7/8	1	1 1/16	1/2	7/16	13/32
9/16	31/32	1 1/8	1 1/8	9/16	31/64	15/32
5/8	1 1/32	1 5/32	1 3/16	5/8	17/32	17/32
3/4	1 1/4	1 7/16	1 1/2	3/4	5/8	5/8
7/8	1 7/16	1 11/32	1 13/16	7/8	23/32	3/4
1	1 5/8	1 7/8	2 1/16	1	13/16	27/32
1 1/8	1 13/16	2 3/32	2 5/16	1 1/8	29/32	31/32
1 1/4	2	2 5/16	2 9/16	1 1/4	1	1 3/32
1 3/8	2 3/16	2 17/32	2 13/16	1 3/8	1 1/32	1 5/16
1 1/2	2 3/8	2 3/4	3 1/32	1 1/2	1 1/16	1 9/32
1 5/8	2 9/16	2 31/32	3 5/16	1 5/8	1 3/32	1 13/32
1 3/4	2 3/4	3 3/16	3 7/16	1 3/4	1 3/8	1 1/2
1 7/8	2 15/16	3 11/32	3 11/16	1 7/8	1 11/32	1 5/8
2	3 1/8	3 5/8	4 1/16	2	1 9/16	1 3/4
2 1/4	3 1/2	4 1/16	4 7/16	2 1/4	1 3/4	1 29/32
2 1/2	3 7/8	4 1/2	4 15/16	2 1/2	1 15/16	2 1/16
2 3/4	4 1/4	4 29/32	5 3/8	2 3/4	2 1/8	2 7/32
3	4 5/8	5 3/8	6 1/16	3	2 5/16	2 5/8
3 1/4	5	5 13/16	7 1/16	3 1/4	2 1/2	2 29/32
3 1/2	5 3/8	6 7/16	7 15/16	3 1/2	2 11/16	3 1/32
3 3/4	5 3/4	6 29/32	8 1/16	3 3/4	2 7/8	3 11/32
4	6 1/8	7 3/8	8 11/16	4	3 1/16	3 1/2
4 1/4	6 1/2	7 7/8	9 1/16	4 1/4	3 1/4	3 17/32
4 1/2	6 7/8	7 31/32	9 3/4	4 1/2	3 7/16	4 1/32
4 3/4	7 1/4	8 15/16	10 1/4	4 3/4	3 5/8	4 9/32
5	7 5/8	8 31/32	10 11/16	5	3 13/16	4 1/2
5 1/4	8	9 7/8	11 3/16	5 1/4	4	4 3/4
5 1/2	8 3/8	9 31/32	11 7/8	5 1/2	4 3/16	4 31/32
5 3/4	8 3/4	10 15/16	12 3/8	5 3/4	4 3/8	5 7/32
6	9 1/8	10 31/32	12 15/16	6	4 9/16	5 7/16

sizes are: $H = d - 1/16''$; $F = 1\frac{1}{2}d + 1/16''$; $h = d - 1/16''$; $h_1 = \frac{1\frac{1}{2}d + 1/16''}{2}$.

Exercise 9.—Draw a bolt with a square head, and nut, as shown in Fig. 42. Make the bolt 1″ in diameter. ***Scale full size.***

Construction.—The proportions of heads and nuts will be found in Table 8. The radius (r) is made equal to F and tangent to the top of the nut or head.

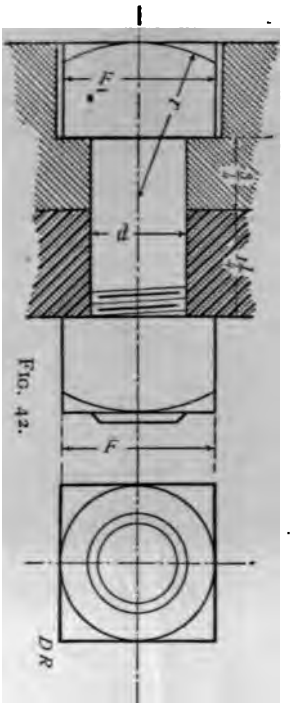


Fig. 42.

A Stud-bolt consists of a bar screwed at both ends (Fig. 43), one end being screwed into the piece upon which the connection is made. The other piece is then passed over the studs and secured by a nut. To allow the nut to make a tight joint, the length of the body or plain part must always be less than the thickness of the piece into which it passes.

Studs are used only when it is impossible, or at least very inconvenient, to use an ordinary bolt. When studs are screwed into cast material, the screwed part should extend into the metal at least $1\frac{1}{4}$ times their diameter, *and*



Fig. 43.

should never be allowed to bear on the bottoms of the holes. Fig. 44 shows a stud used to secure the cylinder-cover (c) to

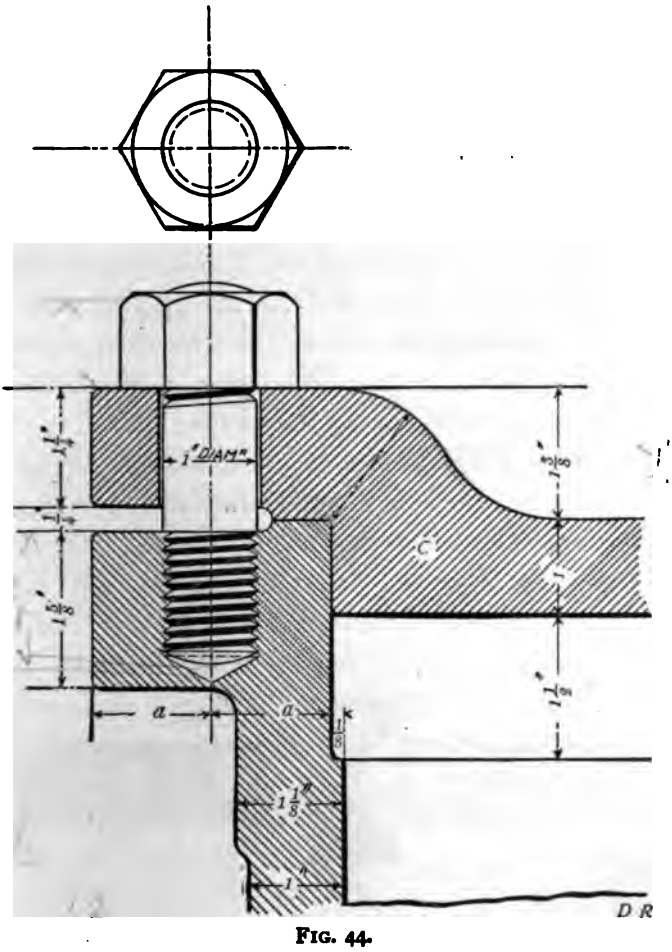


FIG. 44.

the cylinder. Studs are preferred to bolts for this purpose because the flanges can be made very much smaller, and the cover can be removed and replaced without disturbing the cylinder-lagging. A stud should not be placed nearer to the edge of the metal than a distance equal to (d) measured from the centre of the stud, and in steam-tight joints it is usual to make the distance (a) equal to $1\frac{1}{2}d$, as shown in Fig. 44.

Fig. 43 shows the form of stud in general use. The body of this stud is made cylindrical and equal in diameter to the diameter of the screw. As the weakest part of the stud is at the change of section, the form of stud shown in Fig. 44, if subjected to a greater stress than it could withstand, would break off, leaving the screwed part in the metal, but by cutting a semicircular groove of a depth = the depth of the thread on the end of the body that comes in contact with the piece into which the stud is screwed, as in Fig. 43, this part is strengthened and the stud would then break where the upper screwed part joins the body. The broken stud can then be easily removed by means of a pipe-wrench.

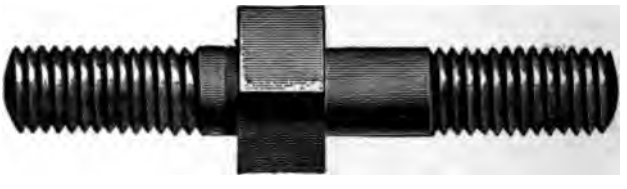


FIG. 45.

In Fig. 45, the stud has a square body which serves as a shoulder, against which the stud may be screwed up tight by means of a wrench applied to the square part. Studs with round bodies are screwed into position by means of a tool

called a stud-nut; this consists of a long nut fitted with an internal screw, as shown in Fig. 46. To avoid damaging the

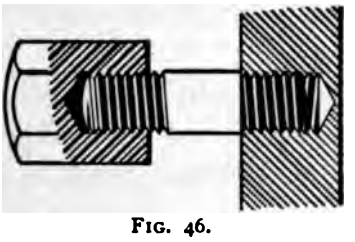


FIG. 46.

point of the stud, the bottom of the screw in the stud-nut is lined with copper. By applying a wrench to the stud-nut, the stud can be screwed into the tapped hole in the metal until stopped by the plain portion on the stud. The stud-nut can then be removed by a quick turn back.

Exercise 10.—Draw a section of a steam-cylinder end-flange, showing the method of securing the cylinder-cover or head (*c*) to the cylinder (Fig. 44). *Scale full size.*

When a bolt-head is of such form, or in a position in which it cannot be held with a wrench to keep it from revolving when screwing up the nut, the bolt is provided with some

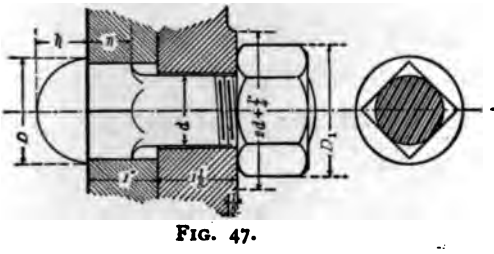


FIG. 47.

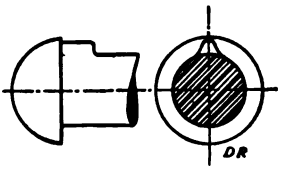


FIG. 48.

device in the body to overcome the difficulty. The spherical or button-headed bolt, shown in Fig. 47, is provided with a square part under the head, which fits into a corresponding hole in the material through which it passes. Another de-

sign used for the same purpose is shown in Fig. 48; this is called a snug, and consists of one or two projections forged on the neck of the bolt and made to fit a correspondingly shaped hole in the metal.

Fig. 49 shows a bolt with a countersunk head and nut. The bolt is kept from revolving by a pin (p), which is driven into a hole drilled in the body of the bolt close to the head. The projecting part of the pin fits into a recess cut to receive it. The nut is provided with holes to receive the spanner used in screwing it up, and may be made equal in diameter to half the amount of metal between the bottoms of the threads

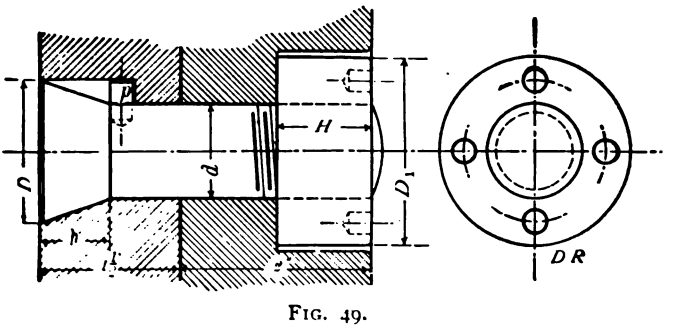


FIG. 49.

and the outside of the nut. The depth of the holes may be made .25 of H, the height of the nut. The projecting part of the pin (p) is usually made square and equal to $.25d$. The pin (p) is sometimes screwed into the bolt to avoid its being lost when the bolt is withdrawn.

The T-headed Bolt shown in Fig. 50 has the sides of the head level with the square neck or body of the bolt, and is used where there is not sufficient room to use bolts of the hexagonal or square-headed form. A common application of this form of bolt is shown in Fig. 50.

The Tap-bolt shown in Fig. 51 makes a fastening without the use of a nut. The bolt is screwed into a tapped hole in one of the pieces to be connected, while the head

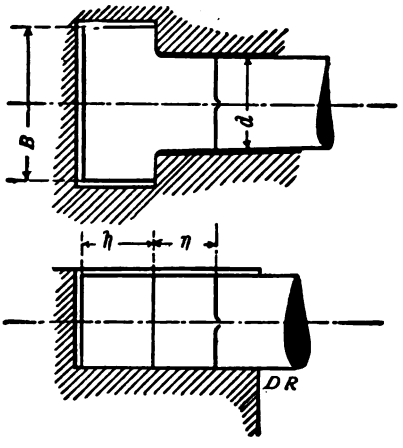


FIG. 50.

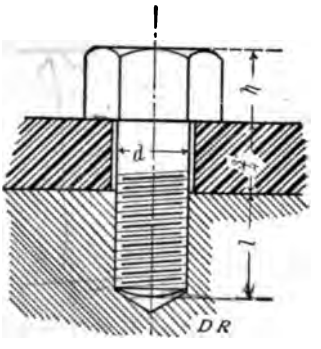


FIG. 51.

presses on the other piece. This form of bolt is used in place of a stud where the piece to be connected could not, if studs were used, be passed over the projecting studs, as in a pipe-fastening where two of the faces are at an angle to each other.

There is no standard for the foregoing bolt-heads and nuts, but the following proportions are in general use:

$$D = 1.5d, \quad D_1 = (1\tfrac{1}{2}d + \tfrac{1}{8}) \times 1.155, \quad B = \text{from } 1\tfrac{1}{2}d + \tfrac{1}{8}'' \text{ to } 2d,$$
$$h = .7d, \quad H = d, \quad n = .6d, \quad l = 1\tfrac{1}{2}d.$$

Exercise 11.—Draw a *spherical* or *button-headed bolt* with a square neck; and a head with a snug on the neck, as shown in Figs. 47 and 48. A *countersunk-headed bolt* with a *countersunk nut* as shown in Fig. 49, a *T-headed bolt* with a square

neck, as shown in Fig. 50, and a *tap-bolt*, as shown in Fig. 51. Make d in each case $= 1''$. *Scale full size.*

Hook-bolt.—This form of bolt is used where it is impossible or undesirable to have bolt-holes through one of the connected pieces. A common application of this bolt is fastening pieces (such as hangers) to flanged beams, as shown in Fig. 52. To keep the bolt from turning, the body is

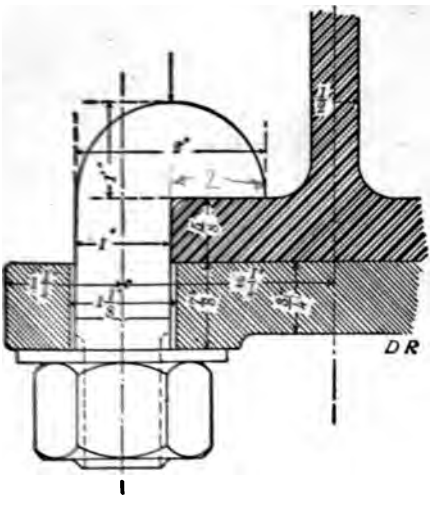


FIG. 52.

made square in cross-section and passes into a correspondingly shaped hole in the connected piece. The diameter of the screw is equal to the square body.

Exercise 12.—Draw an ELEVATION of a hook-bolt, fastening a piece to a flanged beam, as shown in Fig. 52, and PLAN of the bolt only, looking down on the bolt head. *Scale full size.*

Tapered Bolts are used to facilitate fitting where it is necessary that the bolt should be a perfect fit in the hole. Fig. 53 shows a tapered bolt that is in common use in the couplings of propeller-shafts of steamships. As coupling-bolts have only to resist the shearing force, caused by the twisting strain on the shaft, the diameter of the bolt is

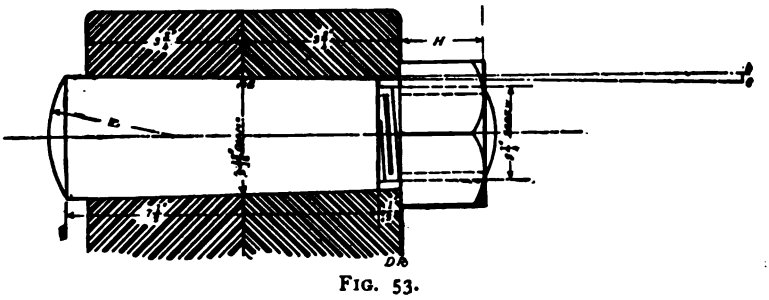


FIG. 53.

the diameter on the line where the two flanges come together, and its strength is equal to $\frac{\pi d^2}{4} f_s$.

As the screwed part of the bolt has only to resist the tension due to screwing up, this part is made smaller in diameter than the small end of the tapered part. In practice, the diameter of the screwed part is generally made equal to $\frac{7d + 1}{9}$, and the height of the nut from $\frac{1}{4}''$ to $\frac{1}{8}''$ less than the diameter of the screw. The advantages gained by using tapered instead of parallel bolts for couplings are: they can be made a perfect fit in the hole, which insures that the different lengths of shaft are in better alignment, are easier withdrawn, and, owing to the diameter of the screw being much smaller than the diameter at the junction of the shafts (i.e., the effective diameter), the flanges can be made smaller.

Exercise 13.—Draw a tapered bolt for a marine shaft-

coupling, showing a part of the shaft-flanges, to the dimensions given in Fig. 53. *Scale half size.*

Construction.—Draw the centre line of the bolt, then the line showing the junction of the flanges, and on this line mark off the diameter of the bolt. From the point (*a*) draw the line *ab* 12 inches long and parallel to the axis of the bolt, and from *b* draw *bc* perpendicular to *ab* and $\frac{1}{16}''$ long, join *ac* which makes the required taper. The radius (*r*) is equal to the diameter of the bolt at the large end.

Exercise 14.—Draw a tapered bolt as in the preceding exercise, leaving off the parts of the shafts, and making the diameter of the bolt 3 inches, and the length of the body equal to 8 inches. *Scale half size.*

Foundation-bolts.—This class of bolts is employed for fastening engine- and machine-frames to stone, brick, or concrete foundations.

The Rag-bolt (Fig. 54).—This form of bolt is fastened to stone by cutting a Lewis hole, which increases in size as it descends. The small end of the hole is made from $\frac{1}{4}''$ to $\frac{1}{2}''$ larger than the large end of the bolt-head. After the bolt-head is placed in the hole, the space between it and the sides of the hole is filled with molten lead or sulphur, thus securing the bolt firmly in position. The frame is cast with a projecting foot through which a hole is cored. This foot passes over the foundation-bolt and the engine- or machine-frame is held in position by the pressure of the nut. The diameter of the hole through the foot is $= d + \frac{1}{4}''$. The diameter of the washer *w* is equal to $2d + \frac{1}{2}''$, and the thickness $\frac{3}{10}$ of *d*. The distance *a* is = half the diameter of the washer $+ \frac{5}{8}''$. The section of the bolt-head is oblong and

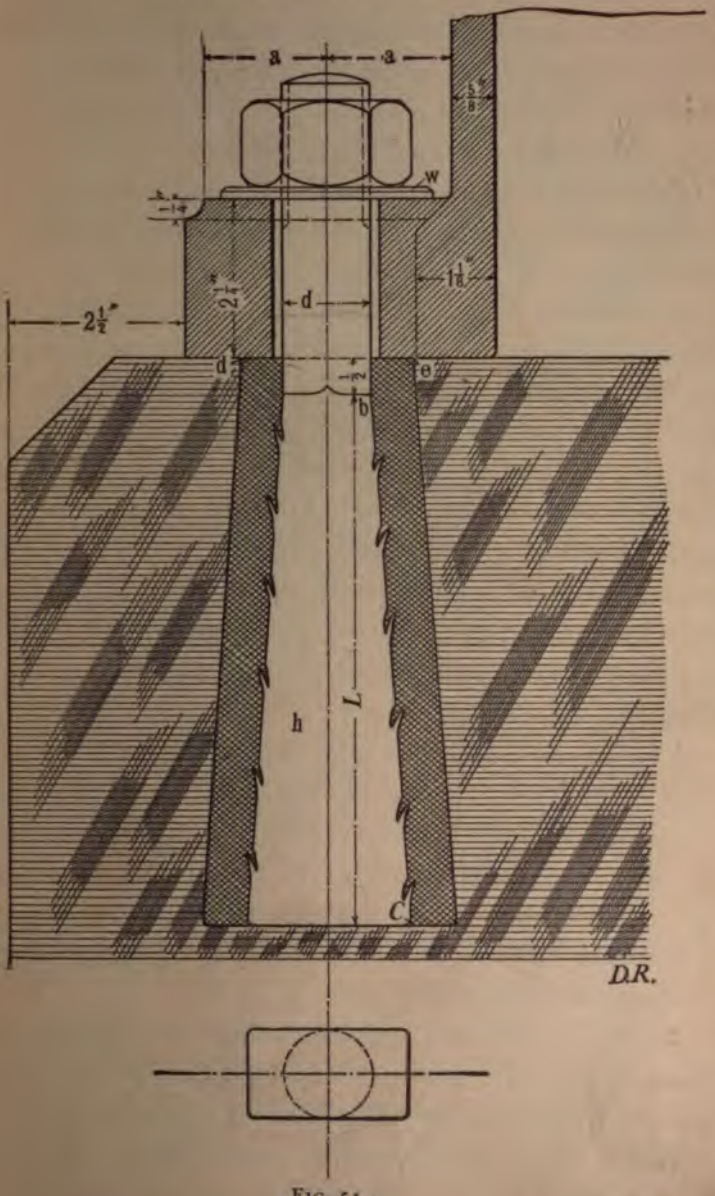


FIG. 54.

purposely made rough and jagged, which obviously increases the resistance the bolt offers against being withdrawn from the hole. The length L of the head (h) is usually made equal to $6d$ and has a taper = $1\frac{1}{2}''$ per foot.

Exercise 15.—Draw a rag-bolt in elevation and plan with a part of a cast-iron engine-frame as shown in Fig. 54, making (d) = $1\frac{1}{4}''$ in diameter. *Scale full size.*

Construction.—Draw the centre line and the line representing the top of the stone foundation, then mark off to (b) the distance which the beginning of the head is below the level of the top of the foundation, and from the point (b) find the taper on one side of the axis in the same manner as in Exercise 13. Make the top of the hole de $\frac{1}{4}''$ greater than the large end of the bolt-head, and through (e) draw a line parallel to the side of the bolt-head bc, which will represent the edge of the hole. To complete the other side of the bolt-head mark off with the dividers equal distances on the other side of the centre line.

The Lewis Bolt, shown in Fig. 55, is used, in some cases, in preference to the rag-bolt, because it can be much more easily removed, which is accomplished by withdrawing the key K. The side bc of the bolt-head (h) has a taper of $1\frac{1}{4}''$ per foot, while the opposite side is parallel to the axis of the bolt. The length L of the head may be made as in the design of the rag-bolt, equal to $6d$.

In Fig. 55, the bolt is shown holding down the pedestal shown in Fig. 54, page 79. The hole that the bolt passes through is rectangular, to allow the pedestal to move laterally. The proportions of the washer are the same as in the last exercise. The thickness (t) of the key is made

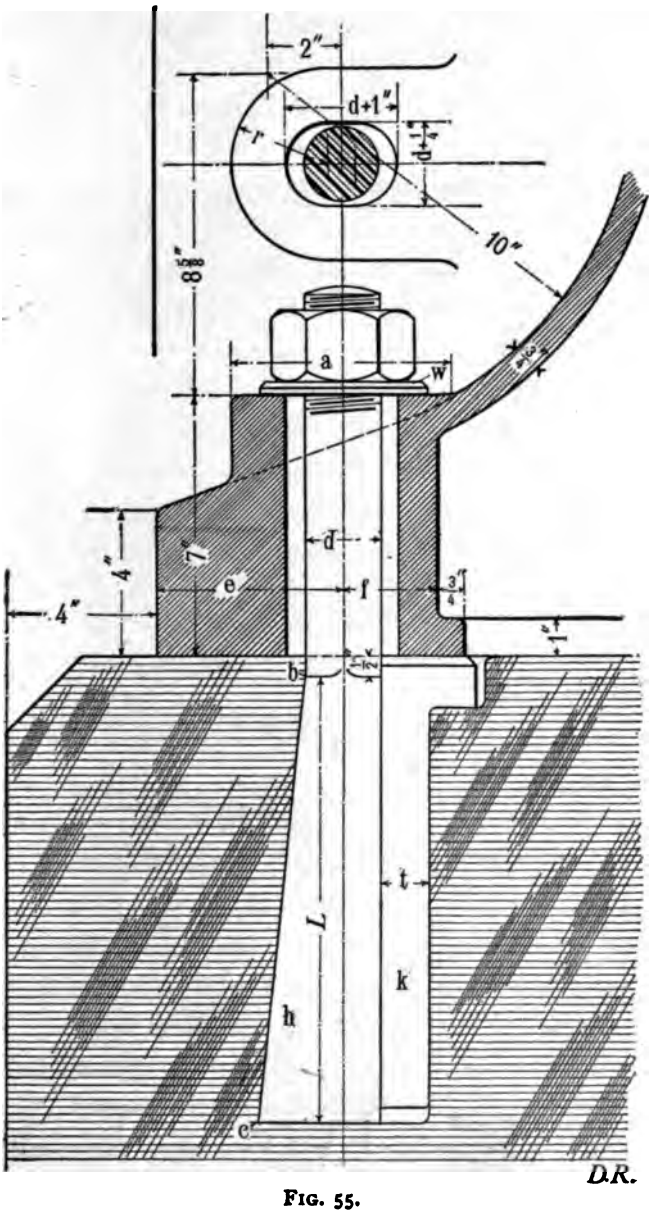


FIG. 55.

sufficient to allow the large end of the bolt-head to pass through the small end of the hole $+\frac{1}{4}''$ for clearance, and the point should stop from $\frac{1}{4}''$ to $\frac{1}{8}''$ up from the bottom of the hole. The length of the key-head is made equal to $2t$, and its thickness equal to t.

Exercise 16.—Draw an ELEVATION of a Lewis bolt showing the method of securing it to the foundation, a section of pedestal-base and a PLAN showing the shape of the hole through which the bolt passes, as in Fig. 55. Draw also an END VIEW of the bolt leaving out the foundation-stone and pedestal-base. Make $d = 2''$ in diameter. *Scale half size.*

Construction.—Proceed in the same manner as in the previous exercise. The distance (a) in this case should be equal to the diameter of the washer (w) + the longitudinal movement $+\frac{1}{8}''$. Make $e = \frac{a}{2} + 2''$, $f = d +$ half the longitudinal movement, $r = \frac{w}{2} + \frac{1}{4}''$.

Anchor-bolts passing through the foundation are recommended in preference to the rag or Lewis bolts wherever it is possible to use them. The heads are made removable, so that the bolts can be inserted from the top, and are either under the foundation or in a recess on the side, as in Fig. 56. The simplest form of removable head is made by screwing a nut upon the lower end after the bolt is in position and driving a split pin through it to keep it from working loose. The objection to this form of head, however, is that the nut cannot be removed without difficulty after it has been in place long enough to rust. The usual and most suitable form of removable head for this class of foundation-bolt is

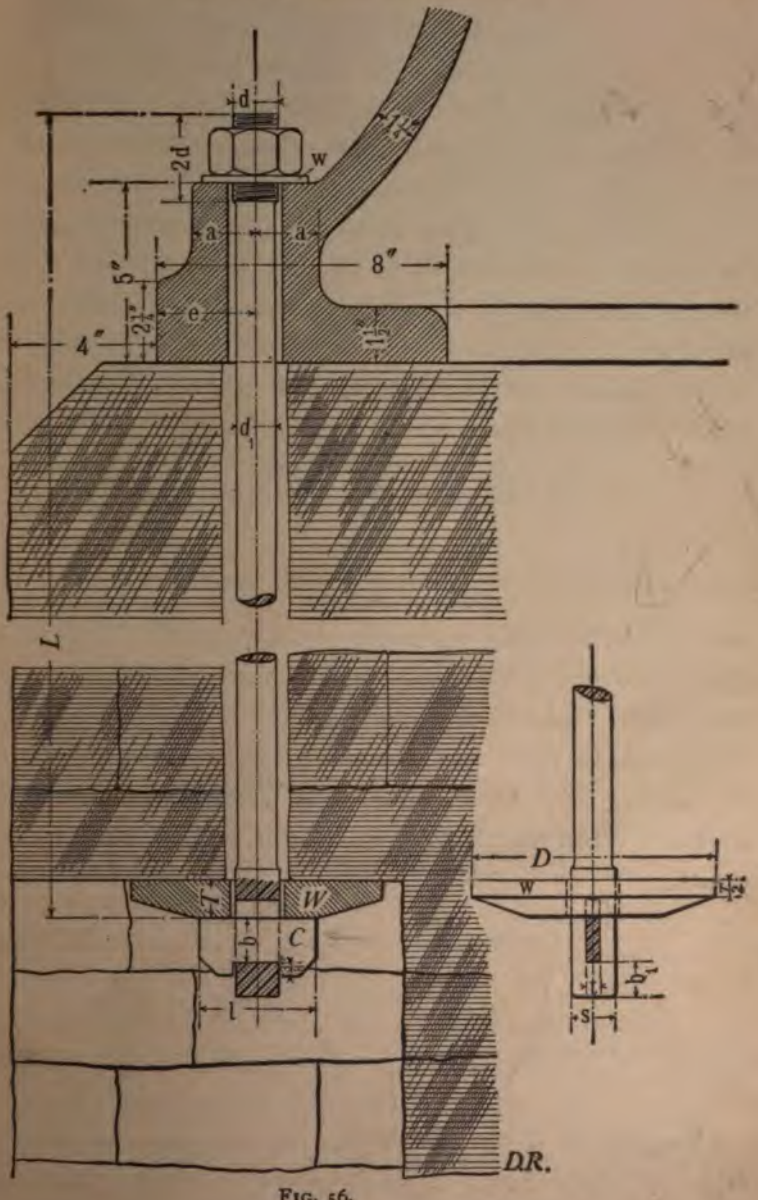


FIG. 56.

shown in Fig. 56. In this design the head end is made square in section, and has a rectangular hole into which the cotter C is fitted. The bolt is kept from turning when the nut is being screwed up by the square end fitting into a corresponding hole in the washer W. To keep the cotter C from working out of place it is provided with gib-heads at the ends. As the strength of a bolt in tension is due to the area at the bottom of the thread, the body of the bolt may be reduced to this extent without reducing its strength.

The proportions of the cotter and the bolt-end through which the cotter passes are

$$b = 1.44d_1, \quad t = .363d_1, \quad S = 1.08d_1,$$

b_1 for shear would $= \frac{d_1}{2}$, but owing to the uncertainty of the longitudinal shearing resistance of the material, it is usual in practice to make it equal S, which insures ample strength. The length l of the cotter should not be less than $2S + \frac{1}{4}''$ and is usually made $= 3S$, which gives a better support to the washer W. The washer W is usually made round or square. When round, D, the diameter, will be found by the formula

$$\frac{\pi D^2}{4} f_c = \frac{\pi d_1^2}{4} f_t, \quad \ldots \quad (12)$$

D may be taken as $= \sqrt{\frac{d_1^2 f_t}{f_c}} - S + \frac{1}{4}''$.

The value of f_c may be taken $= 4$ tons and allow a factor of safety of 20. Take the value of f_t in this case $= 7$

tons per square inch. The thickness T of the washer is made from d, to $1\frac{1}{4}d$.

Exercise 17.—Draw an ELEVATION of an anchor-bolt for securing an engine-frame to a stone foundation showing the frame-foot and stonework in section; the top stone of sandstone and the under part of brick. Make also an END VIEW of the bolt-head with the cotter in section. Make $d = 1\frac{1}{4}''$, $L = 6'\ 0''$, $a = d + \frac{1}{2}''$, $e = 2d + \frac{1}{4}''$. *Scale half size.*

Cap-screws, the different forms of which are shown in Fig. 57, are employed like the tap-bolt for screwing two or more pieces together. The reason for the name "cap-screws" is that they are used for fastening on caps or covers on machinery, such as the caps of journal-bearings, etc.

The Length of a cap-screw is the distance under the head, excepting the flat-headed form, which includes the thickness of the head in the length. The angle of the cone of the flat-headed screw is about 76°, the sides making angles of 52° with the top, but it is usual to represent the heads on the drawings with the sides making an angle of 60° with the top.

The height of the flat-headed screw is = .7 of the screw diameter. The height of the button-headed screw = .6 of the screw diameter. The width of the saw-cuts on the heads are = .2 of the screw diameter. The other proportions are given in Table 9.

Collar-screws are used for the same purpose as cap-screws. The collar under the head, Fig. 58, gives a larger bearing-surface for the head and is used where the hole through the connected piece is larger than the screw diameter.

TABLE 9.

CAP-SCREWS.

(Worcester Screw Co.)

Hexagon and Square Heads.

Diameter of screw	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4
Threads to inch	20	18	16	14	12	12	11	10	9	8	7	7

Hexagon Head.

Diameter of head	7/16	1/2	9/16	5/8	3/4	13/16	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2
Length of head	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4

Square Head.

Diameter of head	3/8	7/16	1/2	9/16	5/8	11/16	3/4	7/8	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8
Length of head	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8

Flat, Round, Fillister, and Button Heads.

Diameter of screw	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1
Threads to inch	40	24	20	18	16	14	12	12	11	10	9	8

Flat Head.

Diameter of head	1/4	3/8	15/32	5/8	3/4	13/16	7/8	1	1 1/8	1 3/8

Round and Fillister Heads.

Diameter of head	3/16	1/4	3/8	7/16	9/16	5/8	3/4	13/16	7/8	1	1 1/8	1 1/4
Length of head	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1

Button Head.

Diameter of head	7/32 Full	5/16	7/16	9/16	5/8	3/4	13/16	15/16	1	1 1/4

Cap Screws.

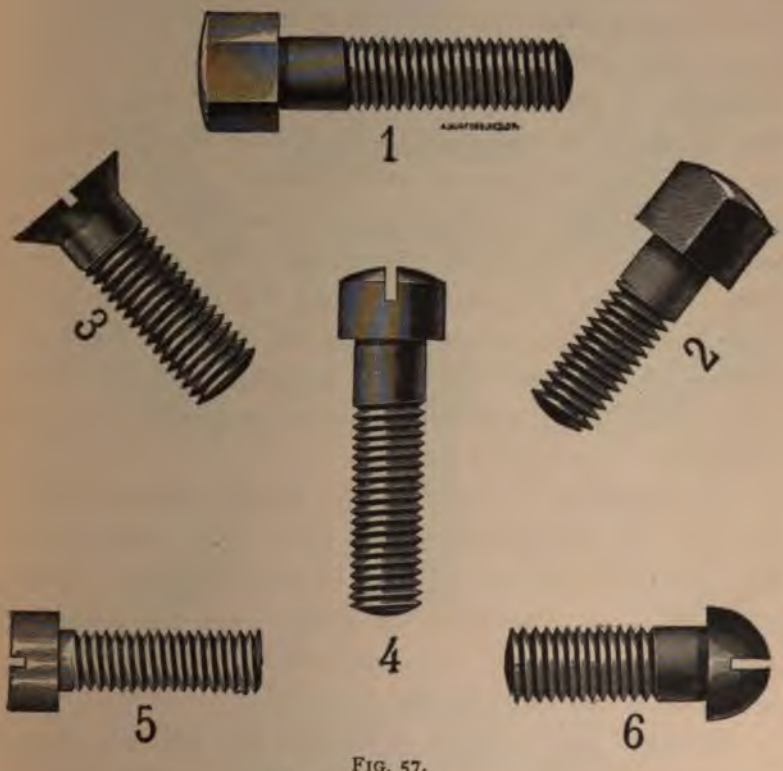


Fig. 57.



Fig. 58.

Set-screws are employed to hold parts of machines in place by setting the point of the screw against the object to be held.

The Holding Power of Set-screws.—In tests (made by G. Lansa, A.S.M.E.) of the holding power of set-screws for securing pulleys to steel shafts by means of wrought-iron screws with points of the form shown at Nos. 1, 2, and 3 (Fig. 59). It was found that the round-pointed form (No. 1), with the radius of the point equal to about the diameter of the screw, had the greatest holding power. The cup-point (No. 2), which was case-hardened, held well while the edges were sharp, but the holding power decreased after the first test because of the edges becoming flat. This serves to show that to get good results with this form of point the screw must be made of a harder material than that of the piece it is holding, and should not be used where the point is subjected to excessive wear.

The length of the head and the distance across the flats is equal to the diameter of the screw. The diameters of the round and flat pivot-points (No. 8) are equal to the diameter at the bottom of the threads, and the length of point = from $.5d$ to d. The angle of the cone and hanger set-points is usually 45° or 60°.

Strength of Bolts.—In an ordinary bolt with a V thread employed for holding two or more pieces together by the pressure due to the screwing up of the nut, the bolt would yield (1) by tension combined with a torsional stress due to the friction between the threads of the nut and those of the bolt. This combination of tension and torsion causes the bolt to part where the thread ends, because of the rapid change of section; (2) by shearing off the threads; (3) by shearing off the bolt-head. Comparing (1) with (2) it will be found that the shearing strength of the thread on the nut is

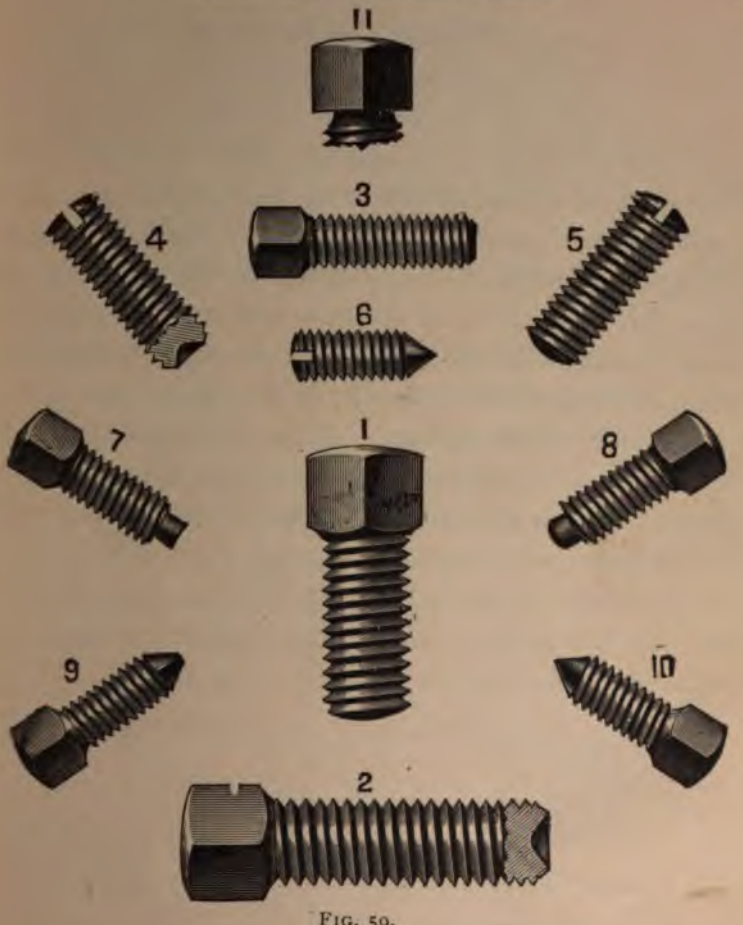


FIG. 59.

SET-SCREWS.

No. 1. Regular round point, set.
" 2. Cup point, set.
" 3. Flat " "
" 4. Cup " headless.
" 5. Round point, headless.
" 6. Cone " "
No. 7. Flat, pivot point.
" 8. Round, " "
" 9. Hanger, set "
" 10. Cone point.
" 11. Necked style.

Diameter of screw	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1	1⅛	1¼
Threads to inch	20	18	16	14	12	12	11	10	9	8	7	7

equal to about twice the strength of the section at the bottom of the thread, but in practice it is found that when the depth of the nut is made less than .7 of the bolt diameter, the threads are injured. Bolts or studs used for face-joints on vessels subjected to internal pressure, depend upon the care exercised by the workman to leave sufficient strength to withstand the pressure after the bolt is screwed up. As the amount of strength left is an unknown and uncertain quantity, the stress upon the bolts calculated from the internal pressure should be kept very low, and no face-joints, unless very small ones, should have bolts less than $\frac{5}{8}''$ in diameter. In permanent joints the stress thus calculated per square inch of section of bolt at the bottom of thread should not exceed 6000 lbs.; and for bolts in joints frequently broken the stress should be as low as 2000 lbs. Thus d_1, the diameter at the bottom of the thread, to withstand the required pressure, will be found by the formula

$$d_1 = \sqrt{\frac{p \times a}{.7854 \times nf_t}}, \quad . \quad . \quad . \quad . \quad . \quad (13)$$

a = area of exposed surface in square inches;

p = the pressure per square inch;

n = the number of bolts;

f_t = the strain per square inch;

Unwin's formula for cylinder-bolts or studs is

$$d_1 = \sqrt{\left(\frac{p}{nf}\right)D}, \quad . \quad . \quad . \quad . \quad . \quad (14)$$

D = the diameter of the cylinder;

p = the pressure per square inch;

n = the number of bolts.

f = the strain per square inch = 2000 lbs. when the diameter of the cylinder is 10″ or less, and 4000 lbs. when above.

Cylinder-cover and steam-chest cover-bolts should be of soft steel.

Bolts of Uniform Strength.—When a bolt in tension is subjected to irregular strains and heavy vibrations, it is made lighter and stronger by making the area of the cross-section of the unscrewed part equal to that of the screwed part at the bottom of the threads. This is obtained by turning down the bolt-body to the same diameter as the screwed part at the bottom of the threads, leaving a part at each end to fit the hole, as shown in Fig. 60.

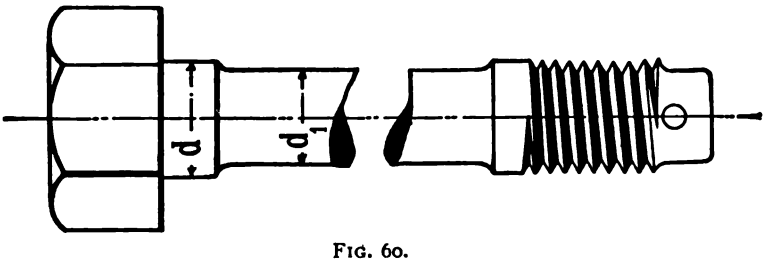


FIG. 60.

Another method adopted where it is necessary that the bolt should fill the hole it is fitted into, is to drill a hole through the centre of the bolt from the head up to where the screw ends, as shown in Fig. 61.

The diameter of the hole is found by the formula

$$d_0 = \sqrt{d^2 - d_1^2}, \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where d = diameter of the bolt;

d' = outside diameter of the bolt-body;

d_1 = the diameter at the bottom of the thread.

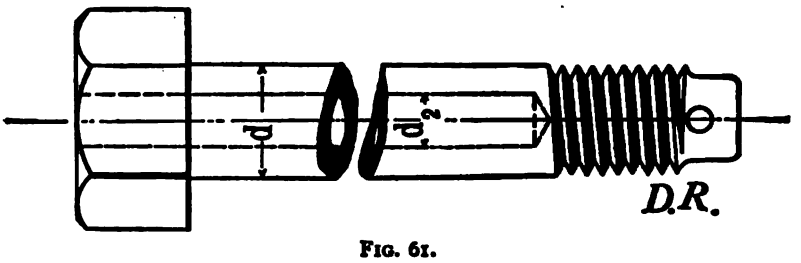


FIG. 61.

Nut-locking Devices.—The pitch of the threads on screw fastenings is such that nuts subjected to constant pressure will not slack back because of the frictional holding power between the threads of the nut and those of the bolt combined with the friction between the bearing-surface of the nut and the piece it is fastening. If, however, the pressure is intermittent and there is much vibration, the nut will slack back when the load on it has been sufficiently reduced to allow the vibrations to overcome the friction which opposes the turning of the nut. Consequently, wherever a screw is subjected to much vibration and a varying load, the nut will gradually slack back and allow the connection to work loose unless some locking device is used to keep the nut from rotating backward.

A Jam-nut is the simplest and most frequently employed device. This is simply a second nut N (Fig. 62) screwed down on the top of the lower nut L as tightly as possible, and the lower nut turned back to cause the threads in the nut N to press upon the under side of the threads on the bolt, while the threads in the nut L press upon the upper

side of the bolt-threads. Hence all slack between the threads of the bolt and those of the nuts is taken up and the nuts will have a frictional holding power independent of the

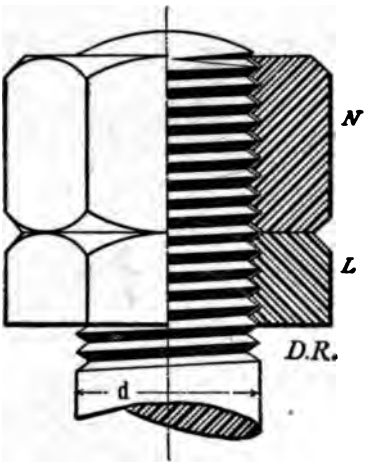


FIG. 62.

tension on the bolt. By this arrangement the load on the bolt is carried on the upper nut, which should be the larger. In practice, however, the thin nut is often put on the top because when screwed down first it requires a special spanner to turn it without disturbing the upper nut, the ordinary spanner or wrench being too thick. The general rule is to make the thin nut equal to half the diameter of the bolt, but many engineers use two ordinary nuts, thus making the height of the nuts equal to twice the diameter of the screw. Others again make a compromise between these methods and make the height of each nut equal to $\frac{3}{4}$ of the screw diameter. We recommend the latter method and have used these proportions wherever jam-nuts are shown. This method of

locking is too cumbersome to be used on large-sized nuts. It is rarely employed on nuts over $1\frac{1}{4}''$ in diameter.

Spring-washer Nut-lock.—This consists of a single coil of a steel spring, *NL*, Fig. 63, which keeps the nut *N* from

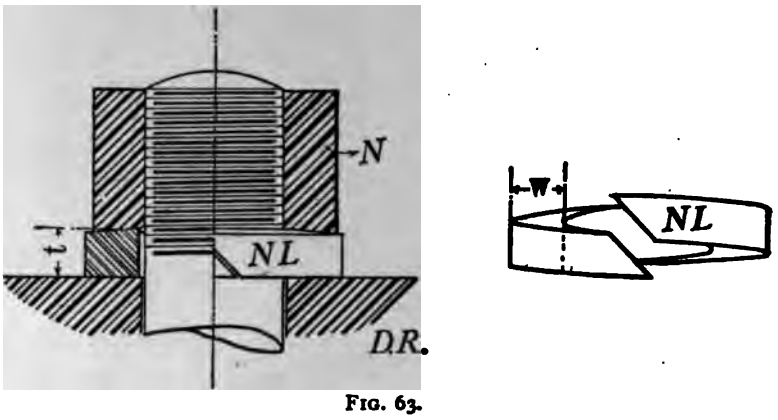


FIG. 63.

slacking back, by its elasticity, when the tension on the bolt is reduced. It is employed quite extensively in railway-engineering practice for securing nuts subjected to the heavy vibrations common to this class of work. The form shown in Fig. 63 is that made by the American Brake Beam Co., and is employed to secure the nuts on the bogie frames, etc., manufactured by them. In the cross-section the top of the washer is inclined $\frac{1}{32}$ of an inch, and when the nut is screwed home its under side conforms to the part of the washer in contact with it. The following proportions agree approximately with the washers manufactured by the afore-mentioned company:

The outside diameter $= F$ the distance across the flats of the nut $+ \frac{1}{16}''$.

The inside diameter = d the diameter of the bolt + $\frac{1}{8}''$.

The mean thickness t is equal to the width w.

Exercise 18.—Draw an elevation of a spring-washer nut-lock before the nut is screwed down, as shown in Fig. 63. Make $d = 1''$ diameter. *Scale twice full size.*

Wiles's Nut-lock, shown in Fig. 64, is an ordinary nut sawn half way across. After the nut is screwed home the

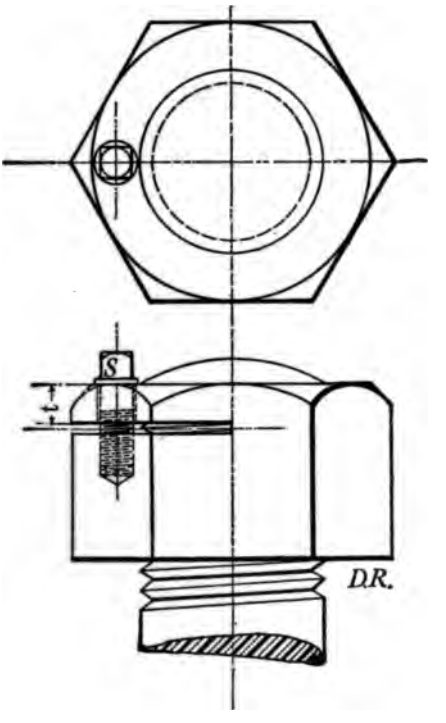


FIG. 64.

opening is partly closed by the screw S, which causes the threads in the upper part of the nut to clamp the corresponding threads of the bolt. The thickness t of the clamping

part of the nut may be made equal to twice the pitch of the threads. The diameter of the screw $S = \frac{F}{2} - \frac{d}{2}$, where F = distance across the flat sides of the nut, and d nominal diameter of bolt. As there is not sufficient room on nuts under 1″ in diameter to use a set-screw, they are locked by partly closing the saw-cut with a hammer blow before the nut is put upon its screw.

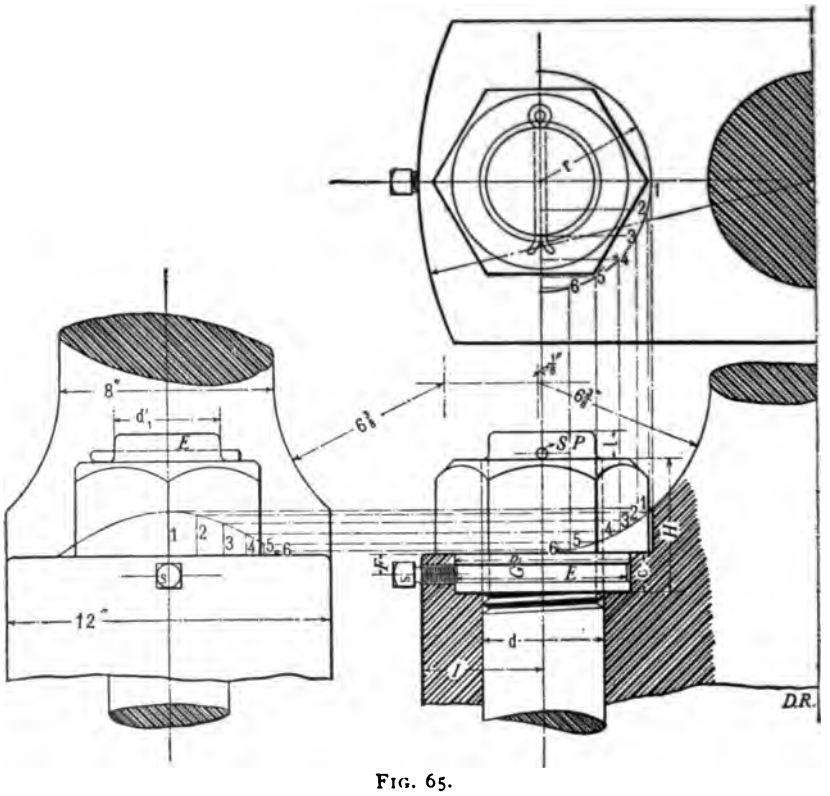


FIG. 65.

Nuts Locked by Means of Set-screws.—The arrangements shown in Fig. 65 are used on quick-moving parts of

machines. They are neat in appearance, simple, and effective when subjected to the worst conditions. In Fig. 65 the lower part of the nut is turned to form a cylindrical projection which fits into a corresponding counterbore in one of the pieces connected by the bolt. Through the latter passes a set-screw S, the point of which presses on the bottom of the groove cut upon the cylindrical projection, to keep the burs raised by the set-screw from interfering with the nut being removed.

The following proportions agree with general practice:

$H = d + \frac{1}{4}''$;
$S = \frac{1}{8}d + \frac{1}{8}''$;
$c = 2S = \frac{1}{4}d + \frac{1}{4}''$;
$F = S$;
$I = d$;
$G =$ diameter of set screw at the bottom of the threads;
$D_1 = 1\frac{1}{2}d$;
$E = 1\frac{1}{2}d - \frac{1}{8}''$;
$r =$ half the distance across the angles of the nut $+ \frac{1}{8}''$.

In addition to the locking device it is usual, on quick-moving parts, to extend the bolt beyond the nut. This extension E, called a pin-point, has the threads cut off and a hole drilled through it into which is fitted a split pin SP. This renders the nut secure against coming off, but does not necessarily prevent its slacking. The diameter of the split pin SP is $.05d + .13$, $l = 2\frac{1}{2}$ times the diameter of the split pin, $d_1 =$ diameter at the bottom of the threads. A method of drawing split pins is shown in Fig. 69.

Exercise 19.—Draw a plan, front elevation and end elevation of the locking arrangement, shown in Fig. 65, showing the application of the arrangement on a connecting-rod end of the form shown in Fig. 66. Make $d = 4\frac{1}{2}''$. *Scale half size.*

Construction.—Locate the centre lines, draw the hexagon and the part of the connecting-rod end in plan. This is as far as we can proceed without the elevation. Draw the part of the connecting-rod in front elevation and complete the locking arrangement, projecting the parts already drawn in the plan view. Taking our measurements from the plan, and projecting from the front elevation, we can complete the end elevation. We can now complete the plan from the front elevation by projecting the parts not already drawn in that view. The method of drawing the curve formed by cutting the fillet to allow the nut to bear upon a flat surface, will be understood by following the construction lines 1, 2, 3, 4, 5, 6. In drawing office practice this curve is usually drawn by an arc of a circle passing through the limiting points. *All parts are dimensioned in inches.*

When it is undesirable to counterbore the piece upon which the nut bears, as in Fig. 65, the cylindrical portion of the nut is made to fit into the collar C, Fig. 66, which is carried upon the outer surface of the connected piece, and kept from rotating by means of the pin P. The nut is secured by the set-screw S passing through the collar and pressing on the bottom of the groove which is cut upon the cylindrical part of the nut. In some cases the pin P is fitted into the hole in the connected piece, but in the example shown in Fig. 66 it is screwed into the piece to avoid the risk of losing it when the nut and collar are removed. The proportions of the nut are the same as in the last exercise. The collar proportions are $D = 2d$, $c = 2S = \frac{1}{4}d + \frac{1}{4}''$. The diameter of $P = \frac{1}{8}d + \frac{1}{16}''$. The length of $P = 2$ times its diameter, half of which fits into the collar.

Exercise 20.—Draw an elevation of the locking arrangement shown in Fig. 66. Make $d = 2''$. *Scale full size.*

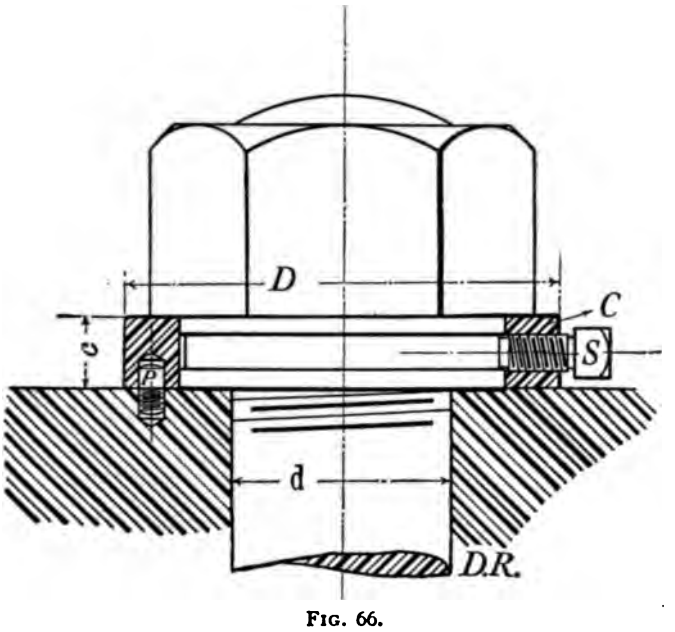


FIG. 66.

Circular Nut-locking Device.—The nut and its locking arrangement shown in Fig. 67 are used for securing the piston-rod to a cross-head of the form shown in Fig. 67. On the outer surface of the nut *N*, longitudinal grooves are cut, into which the projections on the spanner, employed for screwing it up, fit. The locking-plate *LP* consists of a plate shaped to suit the curvature of the nut, and has a projection which fits into one of the spanner grooves. The stud *S* is screwed into the surface upon which the nut is carried, passing through the groove in the locking-plate (*LP*) and is prevented from unscrewing by making the part within the

locking-plate square. The method of locking the nut is as follows: The stud *S* is screwed into the metal at the proper

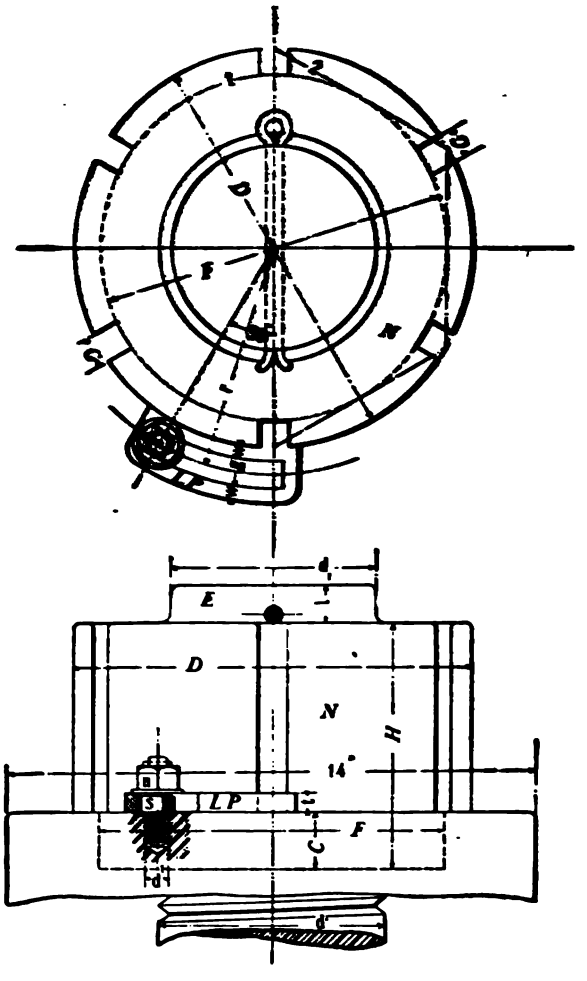


FIG. 67.

distance from the centre of the nut *N*, forming an angle with the radial line which passes through the centre of the projec-

tion on the locking-plate equal to half the angular distance between two of the spanner slots. This allows the nut to be locked in any position. After the nut N is screwed into position, the locking-plate LP, with its projection fitting into one of the grooves, is passed over the stud until it rests upon the piece fastened by the nut N. The nut can then be locked securely by clamping the locking-plate LP, by screwing down N. The nut N has a cylindrical projection on the under side which fits into a corresponding recess in the piece upon which it bears. This insures that the outside of the nut is concentric with the arc which passes through the centre of the locking-plate. It also gives a greater length of nut without increasing the distance which the nut projects from the piece it is fastening.

The proportions of the nut and its locking device are as follows:

D = diameter across the angles of the hexagon;
F = diameter across the flats of the hexagon;
$H = d + 1''$;
$C = \frac{1}{4}d$;
$G = \frac{1}{8}d$;
$t = \frac{1}{10}d$;
$W = .04d + .13$.

Make the diameter d' of the stud $S = \frac{1}{2}''$ when nut N is $2''$ or under, and $\frac{5}{8}''$ for all nuts over $2''$ in diameter. The width of the groove g in the locking-plate equal to the diameter of the stud $+ \frac{1}{16}''$. The width of the square body on the stud $= d'$ and its length $= t - \frac{1}{8}''$.

Exercise 21.—Draw a fluted circular nut and locking

arrangement, as shown in Fig. 67. Make $d = 6''$. *Scale* 8" *to the foot.*

Construction.—Locate the centre lines, draw the circle making the diameter $F = 1\frac{1}{4}d + \frac{1}{4}''$. Tangent to the circle draw the line 2 to make an angle of 30° with the horizontal. Determine the radius r of the arc passing through the centre of the nut-lock and complete the plan of the nut-lock. Determine the centres of the spanner grooves by circumscribing on the half of the circle 1, three sides of a hexagon, as in Fig. 67. The sides of the groove are parallel to the radial lines which bisect the angles formed by the sides of the hexagon. Projecting from the plan complete the elevation. *Construction lines are not to be inked in.*

In Fig. 68 is shown a nut-locking device used for securing the piston-rod to the piston shown in Fig. 66. The nut *N* in this case is of cast steel and has a projection on the under side which fits into a corresponding recess in the locking-plate *LP*, which in turn fits into a circular recess on the piston. The locking-plate has a tapped hole through it, and through this tapped hole, the ring is cut as shown in Fig. 68. After the nut with its locking-plate has been screwed into place a tapered plug *P* is inserted into the tapped hole. This opens the saw-cut and forces the locking-plate against the sides of the circular recess on the piston. The nut is thus securely locked by the friction caused by the pressure of the locking-plate against the sides of the recess. The following proportions may be used for the nut and locking-plate:

d = nominal diameter of screw;

F = distance across the flats—$\frac{1}{8}''$;

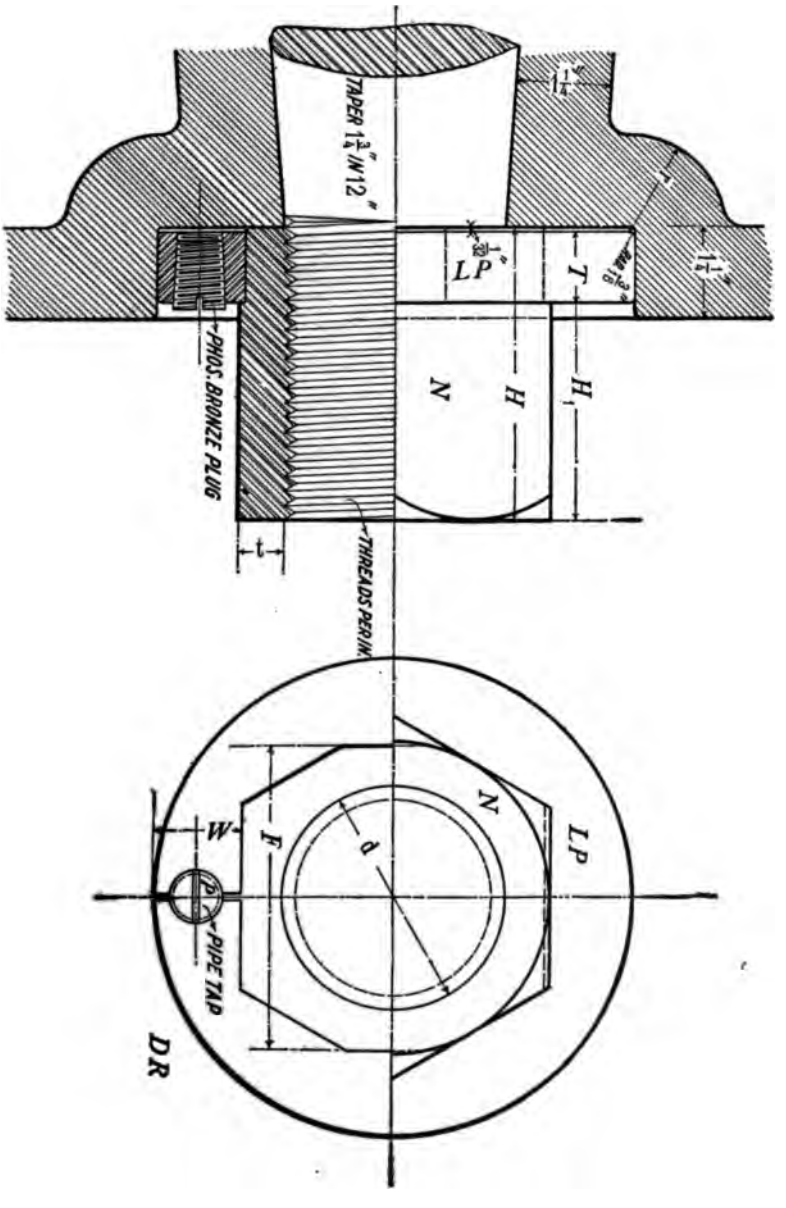

TAPER 1¾" IN 12"
LP
T
N
H
H1
PHOS. BRONZE PLUG
THREADS PER IN.
W
F
d
N
LP
PIPE TAP
DR

FIG. 68.

t = thickness of standard nut having the same number of threads per inch;

$H = d +$ thickness of locking-plate;

$T = .09d + .7$.

The size of the pipe-tap is $= \frac{1}{8}d$, but need not exceed $\frac{8}{8}''$ pipe-tap. The projection on the under side of the nut $= T + \frac{1}{32}''$ to allow the nut to bear upon the piston. $W =$ twice the diameter of the tapped hole at the small end.

Exercise 22.—Draw the nut-locking arrangement shown in Fig. 68, showing part of the piston and piston-rod. Make $d = 3''$ and having 5 threads per inch. *Scale full size.*

Construction.—To find the distance across the flats of the hexagon turn to Table 8, page 68, and find the thickness of a nut having 5 threads per inch by subtracting the radius of the screw from half the distance across the flats. To find the diameter of the tapped hole at the small end, turn to the table of Wrought-iron Pipes on page 57. The size of the actual outside diameter is the diameter of the tapped hole at the large end, and the hole is $\frac{1}{16}''$ less in diameter for every $1''$ of its length. Complete the drawing, substituting the dimensions in inches for the reference letters, and give the number of threads per inch on the piston-rod screw and the nominal diameter of the pipe-tap.

Pin and Pin-joints.—Pins connect pieces by their resistance to shearing at one or two cross-sections.

Split Pins, when made of a uniform diameter from wire of a semicircular cross-section and provided with a head, as in Fig. 69, are used for preventing pieces from separating, while allowing a slight motion in the direction of the axis of the piece that they pass through, as in Fig. 67.

The method of drawing split pins is clearly shown in Fig. 69. The diameter of the pin, in proportion to the diameter d of

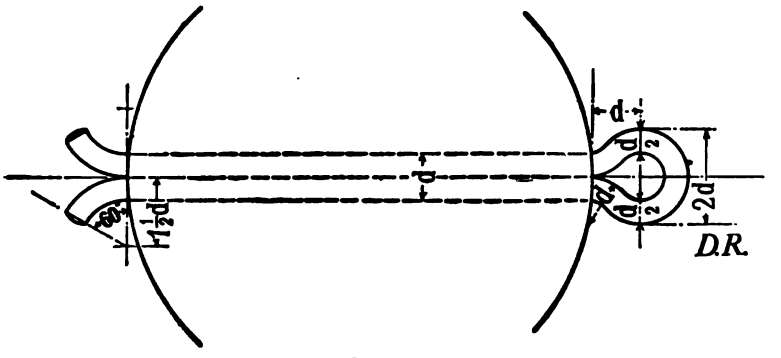


FIG. 69.

the piece it passes through, may be $= .05d + .13$, taking the nearest size in $\frac{1}{16}''$.

Taper Pins, shown in Fig. 70, are used for securing one piece to another in a fixed position, as shown in Fig. 71.



FIG. 70.

They are sometimes split at the small end, and opened out in the same manner as the ordinary split pin, to prevent slacking back. The diameter of the tapered pin at the large end, in proportion to the diameter (d) of the piece through which it passes, may be made $= .06d + .13$ and taking the nearest size from Table 10.

TABLE 10.

STANDARD STEEL TAPER-PINS.

Taper one-quarter inch to the foot.

Number.....	0	1	2	3	4	5	6	7	8	9	10
Diameter at large end	.156	.172	.193	.219	.250	.289	.341	.409	.492	.591	.706
Approximate fractional sizes	5/32	11/64	3/16	7/32	¼	19/64	11/32	13/32	½	19/32	23/32
Longest limit of length	1	1¼	1½	1¾	2	2¼	3¼	3¾	4½	5¼	6

A Knuckle-joint is a pin-joint used for connecting two rods in such a manner that one of them will have a rotary

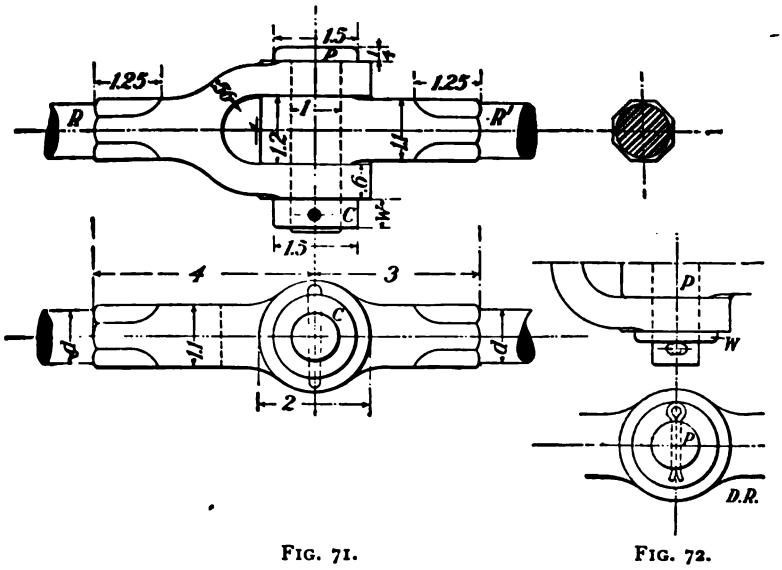


FIG. 71. FIG. 72.

motion in one plane. The connection is made, as shown in Fig. 71, by the pin *P* passing through the fork, or double eye, formed on the rod *R*, and the single eye, on the rod *R'*,

which fits into the fork. The parts of the rods near the eye and fork are either left square or have the corners taken off for a distance, which makes a part of the rod octagonal in cross-section. In the arrangement shown in Fig. 71, the pin P is allowed to turn and is kept in place by the collar C, which is secured to the turning-pin P by driving a taper-pin through it and the collar. The width W of the collar should not be less than $2\frac{1}{2}$ times the diameter of the taper-pin.

Another method in common use for holding the turning-pin in place is to use a loose washer (W) and split pin, as shown in Fig. 72. In Fig. 73, the pin P is held against

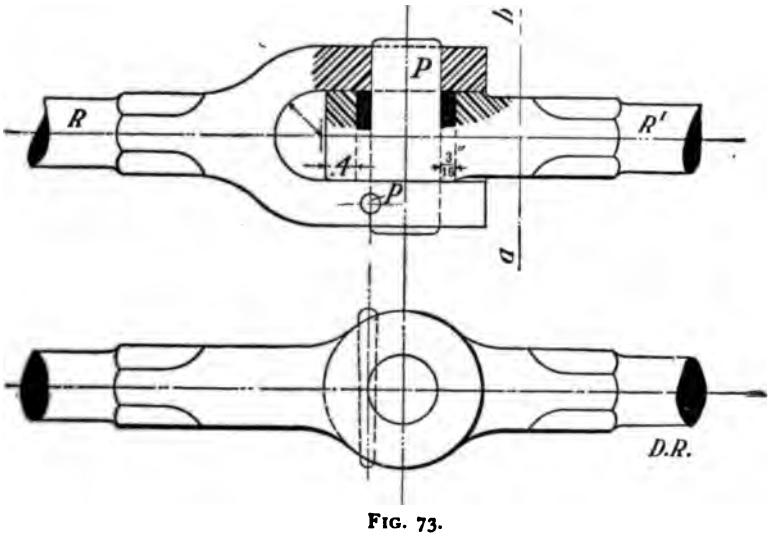


FIG. 73.

turning by a taper-pin p driven transversely through one of the eyes on the rod R and partly into the pin P. By this arrangement all the wear, due to the turning motion, is on the eye of the rod R', which is fitted with a steel or bronze bush.

The Proportions given in Figs. 72 and 73 make joint stronger than the solid rod. This is necessary to a for bending stresses produced when the pin becomes w *Unit of proportions d.*

Exercise 23.—Draw a PLAN, ELEVATION, and END V of the joint shown in Fig. 71, showing the method of hol the pin in place by means of a split pin and washer. M $d = 1''$ *Scale full size.*

Exercise 24.—Draw a PLAN partly in section, an ELE TION and SECTIONAL END VIEW (the plane of section pas through the rod at the line *ab*) of the knuckle joint show Fig. 73. Make $d = 1\frac{1}{2}''$. *Scale full size.*

CHAPTER II.

KEYS, COTTERS, AND GIBS.

Keys are employed to connect wheels, cranks, cams, etc., to shafting transmitting motion by rotation. They are generally made of wrought iron or steel, and are commonly rectangular, square, or round in cross-section. The form of key in general use is made slightly tapered and fits accurately into the key-way, offering a frictional holding power against the keyed piece moving along the shaft. The groove or part where the key fits on the shaft, and the groove into which it fits on the piece it is holding is called the key-bed, key-way or key-seat. For square or rectangular keys, when the keyed piece is stationary on the shaft, the bottom of the groove on the shaft is parallel to the axis, while that of the groove in the piece it is securing is deeper at the one end than the other to accommodate the taper of the key.

Keys may be divided into three classes: 1. Concave or saddle key; 2. flat key; 3. sunk key.

Saddle Key.—This form of key has parallel sides, but is slightly tapered in thickness and is concaved on the under side to suit the shaft, as shown in Fig. 74. As the holding power depends entirely upon the frictional resistance, due to the pressure of the key on the shaft, the saddle key is only

adapted for securing pieces subjected to a light strain. When this key is used for securing a piece permanently, the taper is usually made 1 in 96, but when employed on a piece requiring to be adjusted, such as an eccentric, the taper is increased to 1 in 64 to allow the key to be more easily loosened.

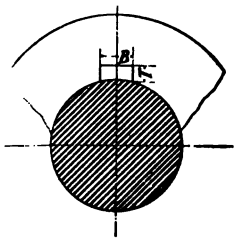


FIG. 74.

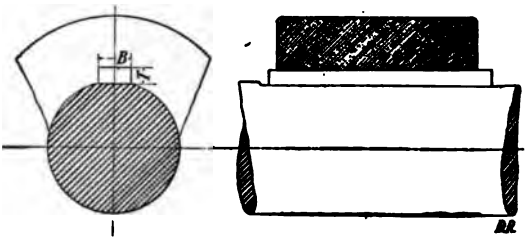


FIG. 75.

Flat Key.—This form of key, Fig. 75, differs from the saddle key in that it rests on a flat surface filed upon the shaft. It makes a fairly efficient fastening, but as it drives by resisting the turning of the shaft under it, there is a tendency to burst the keyed-on piece.

TABLE 11.

DIMENSIONS OF SADDLE AND FLAT KEYS.

D	1	1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
B	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1	1¼	1⅜	1½	1¾
T	3/16	3/16	3/16	¼	¼	5/16	5/16	⅜	⅜	7/16	½	9/16	⅝

Sunk Keys are so called because they are sunk into the shaft and the keyed-on piece, Fig. 76, which entirely prevents slipping. For engine construction they are usually rectangular in cross-section and made to fit the key-seat on all sides When subjected to strains suddenly applied, and

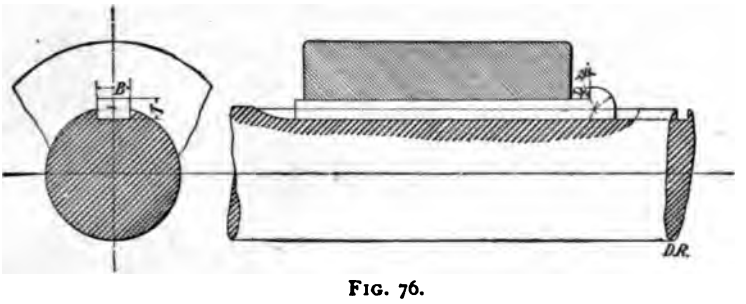


FIG. 76.

in one direction, they are placed to drive as a strut, diagonally, as in Fig. 77.

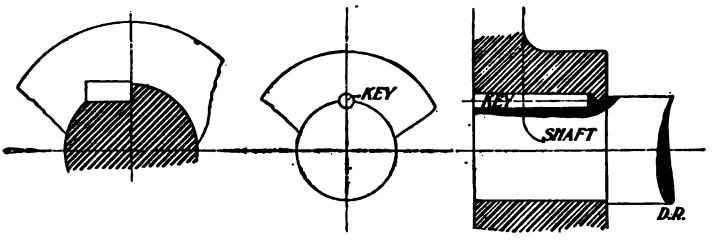


FIG. 77. FIG. 78.

The following table, taken from Richards's "Machine Construction," agrees approximately with average practice:

TABLE 12.

DIMENSIONS OF RECTANGULAR SUNK KEYS.

D	1	1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
B	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1	1⅛	1⅜	1½	1¾
T	5/32	3/16	¼	9/32	5/16	⅜	7/16	½	⅝	11/16	13/16	⅞	1

In mill-work, for fastening pulleys, gear-wheels, couplings, etc., to shafting they are made slightly greater in depth

than breadth. For machine tools they are generally square in cross-section. The following table gives the sizes of keys used by Wm. Sellers & Co. both for shafting and machine tools:

TABLE 13.

	"	"	"	"	"	"	"	"	"
D	1½	1¾	2	2¼	2½	2¾	3	3¼	3½
B	5/16	5/16	7/16	7/16	9/16	11/16	11/16	11/16	11/16
T	⅜	⅜	½	½	⅝	¾	¾	¾	¾
	"	"	"	"	"	"	"	"	"
D	4	4½	5	5½	6	6½	7	7½	8
B	13/16	13/16	13/16	15/16	15/16	15/16	1 1/16	1 1/16	1 1/16
T	⅞	⅞	⅞	1	1	1	1⅛	1⅛	1⅛

Round Keys.—Taper-pins (Fig. 78) are sometimes used as keys to prevent rotation where a crank or wheel is shrunk on to the end of a shaft or axle. Round keys are used in such a case because of the ease in forming the key-way, which is simply a tapered round hole drilled half into the shaft and half into the shrunk-on piece. The standard proportions of the pins are given on page 106. The size at the large end nearest to ¼ of the shaft diameter may be used for this purpose.

Fixed Keys are used when it is undesirable to cut a long key-way on the shaft to allow the key to be driven into place after the keyed-on piece is in position. The fixed key is sunk into the shaft, as in Fig. 79, and the keyed-on piece is driven into position after the key is in place.

When a keyed-on piece has to be adjusted to different positions on the shaft, to avoid the trouble of drawing a tight key in and out, it is made to slide in the key-way, and the keyed-on piece is held against moving along the shaft by means of set-screws, as shown in Fig. 80.

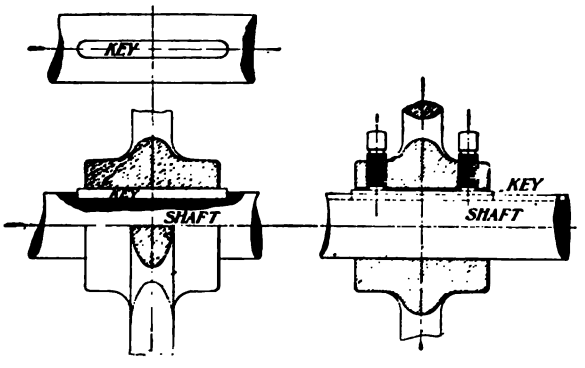


FIG. 79. FIG. 80.

Sliding Feather Key.—This system of keying secures the piece to the shaft, to transmit motion of rotation, and at the same time allows the keyed-on piece to move along the

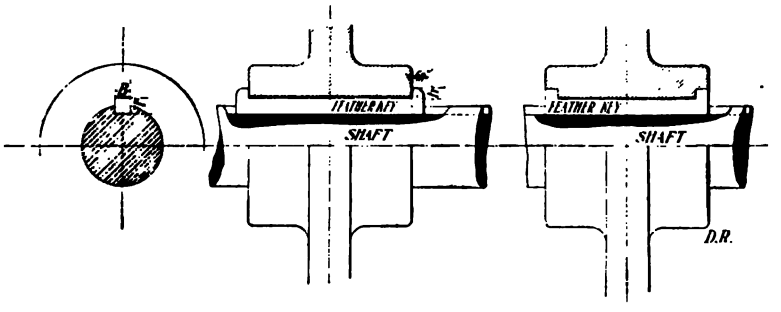


FIG. 81. FIG. 82.

shaft. They may be secured to the keyed piece and slide in a groove on the shaft, as in Fig. 81, or secured to the shaft and slide in the groove in the keyed piece, as in Fig. 79. The dimensions for this form of key may be taken from Table 13.

Woodruff Keys.—This system of keying (Fig. 82*a*) is used for machine tools, or wherever accurate work is of first importance. With this form of key, as the key rights itself to the groove in the keyed-on piece, there is no danger of

the work being thrown out of true by badly fitted keys, and, being deep in the shaft, it cannot turn in the key-seat.

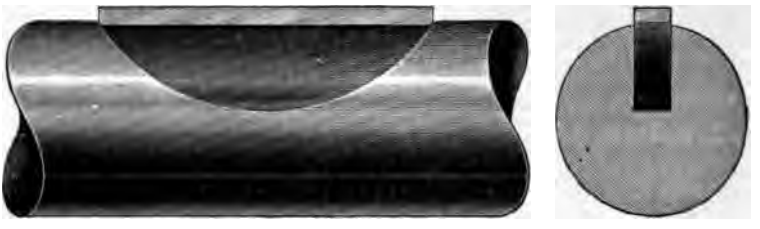


FIG. 82*a*.

Key-heads.—When the point of a key cannot be conveniently reached for the purpose of driving it out, a head is formed on one end, as shown in Fig. 76. Which shows the proportions and method of construction given in RICHARDS'S "MACHINE CONSTRUCTION."

Strength of Keys.—The driving power of saddle keys or keys on flats cannot be calculated with any degree of accuracy. They are used only where the power transmitted by the keyed on piece is small.

Sunk Keys are subjected to shearing and crushing strains, and are required (1) to transmit the whole of the power transmitted by the shaft, as in crank-shaft couplings, etc., or (2) only a part of the power transmitted by the shaft, as when fastening pulleys, eccentrics, etc. As a general rule, however, all keys are proportioned to suit the first conditions, unless where the amount of power transmitted by the shaft is exceedingly great in comparison with that taken off at the keyed-on piece.

Let B = breadth of key;

L = length of key;

$\frac{d}{2}$ = radius at which key offers a resistance;

the shearing of the material which is = 9000 for wrought iron and 11,000 for steel.

$.190d^3f_s$ = modulus of the section of shaft for torsion = $1720d^3$ for wrought-iron and $2182d^3$ for steel shafts;

R = the radius of arm through which P, the power, is transmitted.

Under the first conditions the strength of a tight key would be found by the formula

$$f_s BL\frac{d}{2} = .196d^3f_s, \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

and under the second conditions by the formula

$$f_s BL\frac{d}{2} = PR. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

In the system of sliding keys the crushing action on the key is greater than when the key is a tight fit in the key-way, and keys of this type should be proportioned to have the moment of shaft torsion = the moment of key shearing = moment of key crushing. Then

$$.196d^3f_s = f_s BL\frac{d}{2} = f_c\frac{TL}{2}\frac{d}{2}, \quad . \quad . \quad . \quad (18)$$

and if we take $f_c = 2f_s$, then $T = B$. In practice, however, B is generally greater than T.

Length of Key.—From the foregoing formulæ it will be seen that the strength of the key is directly proportional to (L) the length. To find the length L when the full power of

them in position the taper should not be more than 1 in 24 ($\frac{1}{2}''$ per foot), but where special means are employed for holding the cotter against slacking, the taper may be made as great as 1 in 6 ($2''$ per foot).

Forms and Proportions of Cotter-joints.—When the fastening is subjected to tension only, the arrangement shown in Fig. 84 is used for securing two pieces together by means of a cotter. Fig. 83 shows a method of fastening two rods, R and R', together to resist thrust and tension. The joint is made by fitting the end of the rod R into a socket S formed on the end of the rod R', and through the socket and rod end driving a cotter until the collar C bears against the socket end.

As a cotter-joint is proportioned to withstand the greatest longitudinal force transmitted by the rod, all parts will therefore be proportional to the diameter d_1 of the rod, unless where the dimensions of the rod are increased to insure stiffness. The following proportions are in accordance with good practice:

b, breadth of cotter $= 1.3d_1$;

t, thickness of cotter $= .3d_1$;

d, diameter of pierced rod $= 1.2d_1$;

D, diameter of socket in front of cotter $= 2.4d_1$ or $2d$.

D_1, diameter of socket behind cotter $= 2d_1$;

D_2, diameter of collar on rod $R = 1.5d_1$;

t, thickness of collar on rod $R = \frac{1}{2}d_1$;

l, the length of the rod and socket beyond the cotter = from $\frac{3}{4}d_1$ to d_1.

holes of a diameter equal to the thickness of the cotter and cutting out the metal between them. Again, this form of cotter-way does not weaken the cottered pieces to quite the same extent as when the corners are left sharp. The cotters, however, are not so easily fitted into cotter-ways with round ends, and for that reason some engineers make the cotters of rectangular cross-section, fitted into corresponding cotter-ways.

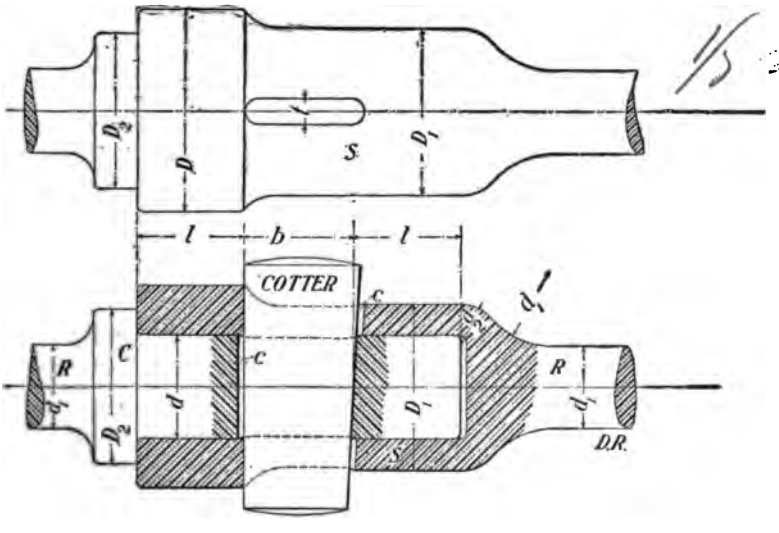


FIG. 83.

Taper of Cotters.—When cotters are employed as a means of adjusting the length of the connected pieces, or for drawing them together, they are made tapered in width, as in Fig. 83, but when used as a holding-piece only, the sides are parallel, as in Fig. 56. When tapered cotters depend upon the friction between their bearing-surfaces for retaining

rod-end may be made from $\frac{1}{8}''$ to $1''$ per foot of length, from 1 in 12 to 1 in 24. The diameter d on the tapered end is taken, when the cotter-way is curved at the where the curve begins, as in Fig. 84, and at the end o cotter-way when the cotter-way is rectangular.

Exercise 25.—Draw a SECTIONAL ELEVATION, a I PLAN, and HALF SECTIONAL PLAN of the cotter-joint sl in Fig. 83. Make $d = 2''$. *Scale full size.*

Exercise 26.—Design cotter-joints suitable for fasten steel piston-rod to the piston and cross-head, as show Figs. 84 and 85. Make the diameter of the rod $d_1 = 2\frac{3}{8}'$ assume that the rod is subjected to a load of 9000 lbs. rod-ends having a taper of 1 in 12. *Scale full size.*

Construction.—Having determined the diameter (d) rod suitable for resisting tensile stresses, then from d fin other proportions of the joints, as in Exercise 25. Me off the distances l and b along the centre line and mark o diameter (d) at the proper point according to the sha the cotter in cross-section, then in the manner given i construction in connection with Exercise 13 draw the end to the given taper. The construction for fi the taper need not be inked in. Complete the dra filling in the actual dimensions and leaving off all refe letters.

COTTER AND GIB.

When one of the pieces connected by the cott a thin strap, as in Fig. 86, a second cotter, call gib, is used. The gib is provided with a head at ends which project over the strap S, thus preventi

(the strap) from being forced open by the friction between it and the cotter as the latter is driven into place. Figs. 86 and 89 show the application of gib and cotter to strap-end connecting-rods, where R is the rod and S the strap. When two gibs are used, as in Fig. 88, the sliding surface on each side of the cotter is the same. Instead of having both gibs tapered, as shown in Fig. 88, one of them may be parallel and the taper all on one side of the cotter. The strength of the gib and cotter in combination is made the same as the

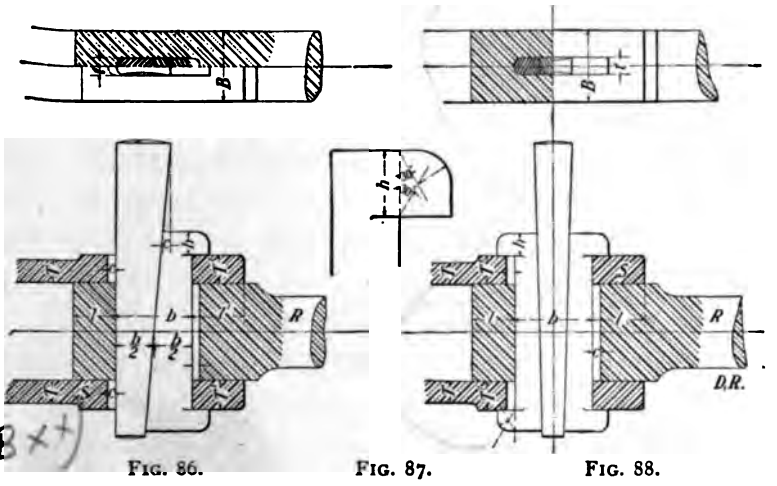


FIG. 86. FIG. 87. FIG. 88.

single cotter and should be proportional to the strap S. The working strength of the strap at the thinnest part is found by the equation

$$2BTf_t = P.$$

from which

$$T = \frac{P}{2Bf_t} \quad \ldots\ldots (20)$$

where P is the maximum pull on the rod, T the thickness,

and B the breadth of the strap. Then as the gib and cotter are to have the same strength as the single cotter, and as B is equal to, or a little greater than d (the diameter of the rod), t may be made equal to $.25B$ and

$$b = 1.3\sqrt{\frac{2BT}{.7854}},$$

T', the thickness of the strap where it is pierced by the cotter, should not be less than $1.3T$. l', the distance from the gib to the end of the strap, $= 2T$. l, the distance from the cotter to the end of the rod, $= 1.5T$. c, the clearance, should not be less than c' (the difference between the widest part of the cotter and the width of the cotter at the top of the gib-head). The method of constructing gib-heads is shown in Fig. 87, where h, the height of the gib-head, $= 1\frac{1}{4}t$.

Cotter-locking Arrangements.—A simple method, and one that is used in nearly all cases, where possible, is to screw one or two set-screws through the rod until the point or points press against the cotter. To keep the burs, raised by the point of the screw, from interfering with the motion of the cotter, the set-screw bears on the bottom of a shallow groove cut on the side of the cotter, as shown in Fig. 89. The diameter of the set-screw need not exceed $\frac{5}{8}''$. The length of the groove is equal to the travel of the cotter + the diameter of the set-screw. The travel of the cotter is the distance from the top of the gib (or where no gib is used, from the top of the piece into which the cotter passes) to the top of the cotter when the cotter is just in place.

The width of the groove is equal to the diameter of the set-screw point, and the depth $= \frac{1}{16}''$.

In Fig. 90 the cotter is locked by an upper and lower nut upon a screwed extension of the gib, which passes through a head formed on the cotter. This arrangement is used for fastening in, and may be used for forcing the cotter into, position.

d, the diameter of the screw $= t$;
h, the height of the head $= 1\frac{1}{4}d$.

As the axis of the locking-screw is not parallel with the side of the cotter that is in contact with the gib, the hole in the cotter head through which the screw passes is elongated

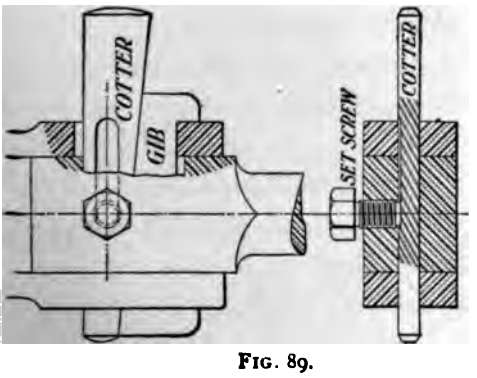


FIG. 89.

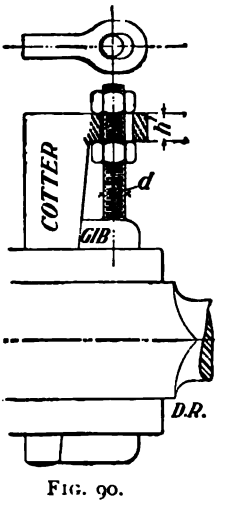


FIG. 90.

to an amount equal to the taper of the cotter in its length of travel $+ \frac{1}{16}''$ for clearance.

Exercise 27.—Draw a SECTIONAL ELEVATION and a HALF SECTIONAL PLAN, a PLAN, and SECTIONAL END VIEW of a gib- and cotter-joint to resist a tension of 12,000 lbs. Make the diameter (d) of the rod $= 2''$. The cotter to have an adjust-

ment of $\frac{3}{8}''$ with a taper = 1 in 8. Show the method of locking the cotter by means of a set-screw. *Scale full size.*

Exercise 28.—Draw a SECTIONAL ELEVATION AND PLAN of a double gib- and cotter-joint as shown in Fig. 88, with the locking arrangement shown in Fig. 90. The joint to be proportioned to resist a tension of 33,000 lbs., the cotter to have an adjustment of $\frac{3}{4}''$ and a taper of 1 in 10. Take the diameter of the rod = 3″. *Scale full size.*

CHAPTER III.

RIVETS AND RIVETED JOINTS.

RIVETS are made from round bars of steel, wrought iron, copper, or brass, and are used to fasten two or more plates permanently together.

The plates to be riveted are either drilled or punched with holes $\frac{1}{16}''$ larger in diameter than that of the rivet-shank.

When the rivet is placed in position through the plates a sufficient length of shank projects beyond the plates to provide for forming the *rivet-point head* either by hammering or by machine-pressure (see Fig. 92). Unwin calls a riveted joint the "simplest permanent fastening."

Rivets are made by being pressed into shape while red-hot with rivet-making machines using dies of suitable size and form.

The names and proportions of rivet-heads shown by Figs. 92 to 96 will be given later.

The end of the rivet opposite to the head before riveting up is called the point and after riveting the point-head.

Just before using the rivet is heated red-hot and when placed in position for hand-riveting is held there by means of a large hammer with a long handle fulcrumed at a convenient distance from the rivet and a man's weight applied at the end, while the point-head is made by two riveters either in the form of the steeple head by hammers only or the snap head (Fig. 92) by using a cup-shaped die called a snap.

In machine-riveting the point-head is pressed into shape by suitable dies, the motive power being either a lever, steam, hydraulic, or pneumatic pressure. Machine-riveting upsets the rivet and fills the hole much better than hand-riveting, because the steady even pressure of the former is exerted uniformly through the whole of the rivet.

Hydraulic riveting is preferred to steam-riveting, because the pressure from the former can be gradually applied, while the force from the latter generally comes upon the rivet with such rapid blows that sufficient time is not allowed for the rivet to properly fill the hole.

Rivet-holes punched through rigid steel plates should always be annealed after punching, because the punching injures the material surrounding the hole to such a dangerous extent that the elasticity of the plate is destroyed, and when the joint is subjected to strain the stress is not uniformly distributed between the rivet-holes. Another difficulty with punched holes is the imperfect spacing of the rivet-holes.

Drilled holes are usually more expensive than punched holes and the sharp square edge is not as favorable to the resistance of the rivet to shearing, but they are more accurate in size and spacing, and the resistance of the rivet to shearing can be increased by slightly rounding the edges of the holes.

Calking.—No riveted joint is ever perfectly steam-tight without calking. This is a process by which a narrow strip of the bevelled edge of one plate is brought into forcible contact with he plate beneath it.

At *a*, Fig. 91, is shown the calking-tool commonly used in hand-calking, and at *b* an improved form of calking-tool patented bv Mr. J. W. Connery of Philadelphia and known as

the concave calking-tool from the concave finish given to the calked edge of the plate. This is a favorite style of calking with locomotive-builders for high-pressure boilers.

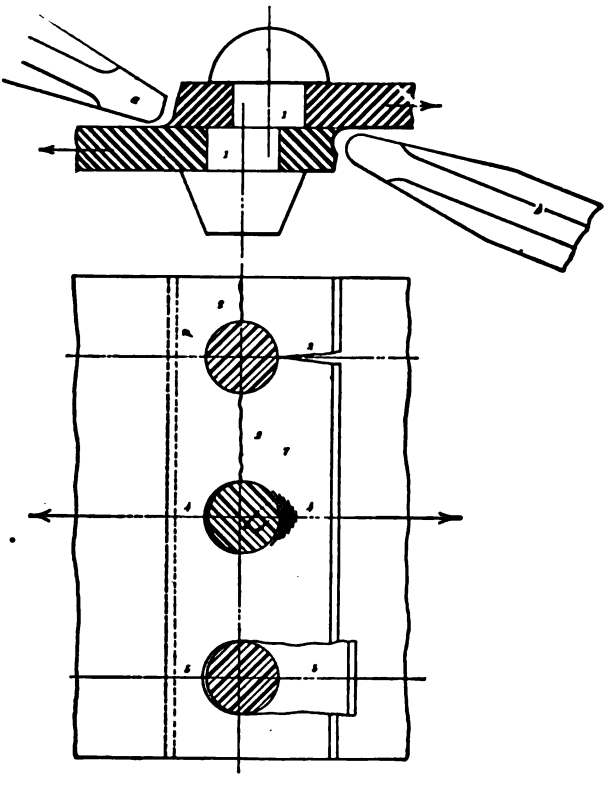


FIG. 91.

Calking with pneumatic calking-hammers has become quite general in most first-class boiler-shops. Peabody and Miller in their "Steam-boilers" describe a pneumatic calking-machine as follows:

"In general principle it resembles a rock-drill and consists of a cylinder in which works a piston and rod on the end of

which is the calking-tool. Air is supplied for working the piston at a pressure of 50 or 60 lbs. through a flexible tube. It makes about 1500 working strokes a minute $\frac{5}{16}$" long. The calker which is about $2\frac{1}{2}$" in diameter outside and 15" long over all, is held by a workman who presses it slowly along the seam to be calked. The edge of the tool is well rounded, so as not to injure the lower plate. Work can be done four times as rapidly with the pneumatic calker as by hand."

The edges of rivet-heads are not calked except when they show a leak during the process of testing. In some of the largest boiler-shops an inspector is employed part of whose duty it is when examining a boiler to discover if any of the rivets are loose. This is done by placing a finger on the under side of the suspected rivet and tapping the top of it with a small hammer made for the purpose; if the rivet is not perfectly tight it will be easily detected by the finger; in such a case the loose rivet is cut out and replaced by a new one.

The Forms of Rivets.—The standard forms of rivets in general use are: (1) the *button* head (Fig. 92); (2) the

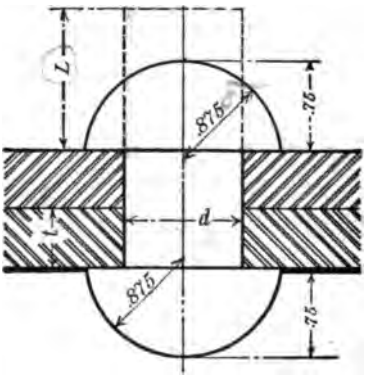


FIG. 92.

conical head (Fig. 93); (3) the *steeple* head (*b*) (Fig. 94); (4) the *steeple* head (*d*) (Fig. 95); (5) the *countersunk* head (Fig. 96).

The *button* head, or, as it is sometimes named, the *snap* head, is usually made with a machine-riveter.

The *conical* is also a machine-formed head and is commonly used with a button point-head or tail and sometimes with a steeple point.

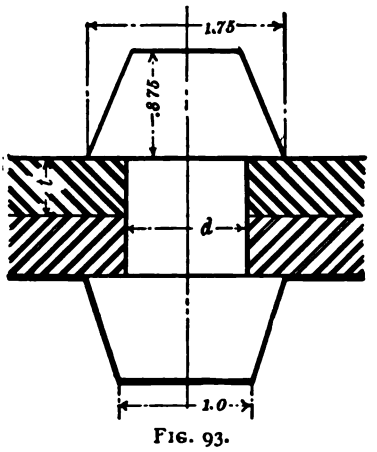


FIG. 93.

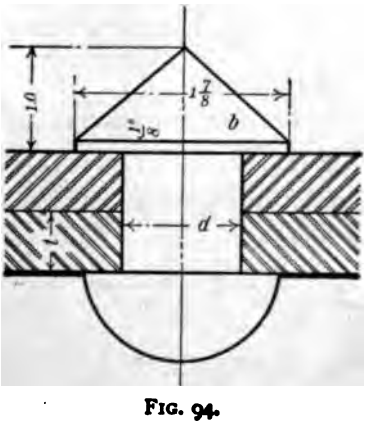


FIG. 94.

The *steeple* point-head is the form mostly used in hand-riveting.

The *countersunk* point-head is only used when there is not sufficient room for one of the other forms and should never be used unless it is impossible to avoid it. It is more costly than, and not as strong as, the other forms.

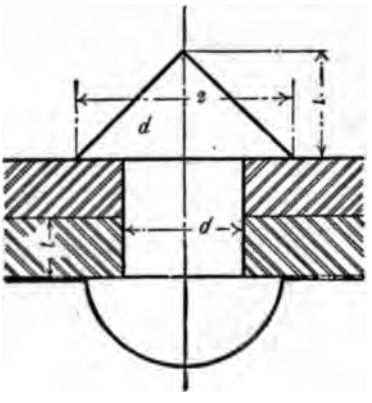


FIG. 95.

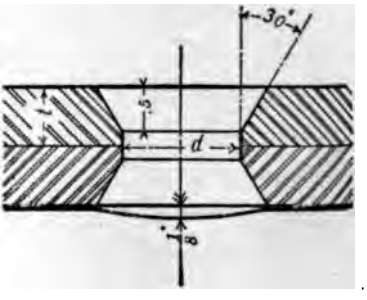


FIG. 96.

Proportions of Rivet-heads.—The proportions given in the figures in terms of the diameter d are those used by the Champion Rivet Co. and agree closely with general practice.

Length of Rivet-shank.

The length L (Fig. 92) for countersunk point-head and 2 plates..... $1d$

For countersunk point-head and 3 plates..... $1d + \frac{1}{8}''$

For steeple point-head..... $1.25d$

For steeple point-head, large, machine-driven..... $1.50d$

For button point-head..... $1.25d$

The above proportions are good for ordinary boiler-plates, but, since the holes are $\frac{1}{16}''$ larger than the rivet, the shank

should be increased in length for *thick* plates to properly fill the additional annular space.

The rivet-shank is usually about $\frac{1}{16}''$ smaller in diameter than the hole and has a slight taper toward the point.

Exercise 29.—Make a drawing of each style of riveting shown in Figs. 92 to 96, making t equal to $\frac{3}{4}''$ and selecting from Table 14, page 135, the diameter of rivet. For conventions see page 22. *Scale full size.*

Riveted Joints.—There are in common use at least five different styles of riveted joints, viz.: the single-riveted lap-joint (Fig. 97); the double-riveted lap-joint with *staggered* spacing (Fig. 98); the double-riveted lap-joint with *chain* spacing (Fig. 99); the single-riveted butt-joint with *chain* spacing (Fig. 100); the double-riveted butt-joint; the multiple-riveted lap-joint which has more than two rows of rivets in the lap; the multiple-riveted butt-joint which has more than two rows of rivets on each side of the line where the plates butt together (Fig. 103).

NOTATION.

d = the diameter of the rivet-hole or of rivet when riveted up.

p = the pitch of the rivets, i.e., the distance from the centre of one rivet to the centre of the next in the same row (Fig. 97).

l = the distance from the centre of rivet-hole to edge of plate (Fig. 97).

r = the distance between the rows on double-riveted joints.

r_1 = the distance between outside rows of rivets on lap-joints with welt-strip and butt-joints.

m = the least distance between the edge of rivet-hole and edge of plate = *margin* (Figs. 97 to 103).

t = the thickness of plate.

t_1 = thickness of outside welt-strips for butt-joint.

t_2 = thickness of inside welt-strips for butt-joint.

t' = thickness of inside welt-strips for lap-joint.

f_t = the tensile strength per square inch of the plate in lbs.

f_s = the shearing strength per square inch of the rivet in lbs.

f_s' = the shearing strength per square inch of the plate in lbs.

f_c = the compressive or crushing strength per square inch of the plate in lbs.

R = the radius of boiler in inches on the outside of course of smallest diameter.

N = the width of widest welt-strip.

K = the width of narrowest welt-strip.

P = working pressure in lbs. per square inch.

D = outside diameter of boiler-shell at course of smallest diameter.

F = factor of safety.

E = efficiency of riveted joint.

T = total tensile stress.

a = area of rivet-hole = $.7854d^2$.

Strength of Single-riveted Joint.—There are five different ways in which a single-riveted lap-joint may give way:

(1) Shearing the rivet, as shown at 1 in Fig. 91.

(2) Tearing plate along the centre line of rivets, shown at 2, 2.

(3) Tearing the plate through the margin, shown at 3.

(4) Crushing the rivet or the plate in front of the rivet 4, 4).

(5) Shearing the plate in front of the rivet (5, 5).

The shearing strength of the rivet

$$= \frac{\pi d^2}{4} \times f_s = a \times 38{,}000. \quad . \quad . \quad . \quad (1)$$

The resistance of plate to tearing on centre line of rivet

$$= (p - d)t \times f_t. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The resistance of the plate to tearing at 3 has been found by experiment to be great enough when the distance l is made equal to $1\frac{1}{4}d$, and, as this rule agrees with general practice, it will be maintained throughout this work.

The compressive resistance of the plate at 4 is

$$t \times d \times f_c. \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The resistance to shearing the plate in front of the rivet as shown at 5, 5.

$$= 2t \times l \times f_s'. \quad . \quad . \quad . \quad . \quad . \quad (4)$$

But if the joint is made strong enough to resist shearing the rivet or tearing the margin it will be strong enough to resist shearing or crushing the plate in front of the rivet, so that the latter may generally be disregarded.

The thickness of the boiler-plate is

$$t = \frac{P \times R \times F}{f_t \times E} = \frac{PR}{11{,}000E}. \quad . \quad . \quad . \quad (5)$$

The value for t should be taken as the nearest even sixteenths of an inch. Take $E = .70$.

The thickness of dome-sheet may be calculated by the same formula.

In locomotive-boilers the thickness of tube-sheets for $\frac{5}{8}''$ shells and over should be $\frac{1}{2}''$ to $\frac{9}{16}''$.

When shells are less than $\frac{5}{8}''$ thick it is usual to make the thickness of tube-sheets equal to $t + \frac{1}{8}''$.

The throat-sheet is usually made $\frac{1}{8}''$ thicker than the shell to allow for extra flanging.

In thick shells, $\frac{3}{4}''$ or over, $\frac{1}{16}''$ thicker will be sufficient.

When the back tube-sheet is separated from the fire-box throat-sheet the latter should be made the same thickness as the fire-box side sheets, viz., $\frac{5}{16}''$.

The fire-box crown-sheet is usually made $\frac{3}{8}''$ and the side and door sheets $\frac{5}{16}''$ thick.

Diameter d of Rivet-hole.—It is very desirable in designing riveted joints to obtain the highest efficiency and still maintain a proper tightness by using a pitch not too long for calking.

In determining the diameter d of the rivet it is necessary that it should be strong enough to resist both shearing and crushing. Now the resistance to shearing is

$$\frac{\pi d^2}{4} f_s,$$

while that of crushing is

$$dtf_c,$$

which shows that the latter increases as the diameter and the former as the square of the diameter. So that if we can obtain such a relation between the length of the pitch and the

diameter of the rivet-hole as will give the highest efficiency consistent with tightness the crushing strength of the rivet or the plate in front of the rivet need not be considered.

To our knowledge the maximum limit for the length of pitch that will insure perfect tightness of the joint has never been ascertained by experiment or test, so that we have to depend largely on existing practice in determining the ratio between d and t.

Mr. Wm. M. Barr in his "Boilers and Furnaces" gives the following ratios between the thickness of the plate and the diameter of the rivet for single-riveted lap-joints, using the nearest even sixteenths of an inch, for steel plates and steel rivets (tensile strength of plates 55,000 lbs. and shearing strength of rivets 44,625 lbs. per square inch):

TABLE 14.

t	Ratio of d to t.	d	Decimal Equivalent.	Area of d. Sq. In.	Pitch of Rivets.	
					Decimal.	Working Fraction.
$\frac{1}{4}''$	2.75	$\frac{11}{16}''$	.6875	.371	1.892	$1\frac{7}{8}''$
$\frac{5}{16}''$	2.40	$\frac{3}{4}''$	.75	.442	1.897	$1\frac{7}{8}''$
$\frac{3}{8}''$	2.17	$\frac{13}{16}''$	.8125	.518	1.934	$1\frac{15}{16}''$
$\frac{7}{16}''$	2.00	$\frac{7}{8}''$	.875	.601	1.990	$2''$
$\frac{1}{2}''$	1.87	$\frac{15}{16}''$	.9375	.690	2.058	$2\frac{1}{16}''$
$\frac{9}{16}''$	1.78	$1''$	1.000	.7854	2.133	$2\frac{1}{8}''$
$\frac{5}{8}''$	1.70	$1\frac{1}{16}''$	1.0625	.887	2.205	$2\frac{3}{16}''$
$\frac{11}{16}''$	1.64	$1\frac{1}{8}''$	1.125	.994	2.298	$2\frac{5}{16}''$
$\frac{3}{4}''$	1.58	$1\frac{3}{16}''$	1.1875	1.108	2.386	$2\frac{3}{8}''$

A committee of the Railway Master Mechanics' Association on riveted joints in 1895 gave the following ratios between d and t in their report for single-riveted lap-joints (steel plates of 55,000 lbs. tensile strength and iron rivets of 38,000 lbs. shearing strength per square inch)

TABLE 15.

t	Ratios.	Mean Ratios.	d	p	E
$\frac{1}{4}''$	2.25 to 3.00	2.62	$\frac{11}{16}''$	$1\frac{3}{8}''$	58.6%
$\frac{5}{16}''$	2.00 to 2.80	2.40	$\frac{3}{4}''$	$1\frac{1}{2}''$	55.8%
$\frac{3}{8}''$	2.00 to 2.60	2.30	$\frac{7}{8}''$	$2''$	55.3%
$\frac{7}{16}''$	1.71 to 2.42	2.06	$1\frac{1}{16}''$	$2\frac{1}{16}''$	52.8%
$\frac{1}{2}''$	1.75 to 2.35	2.05	$1''$	$2\frac{1}{16}''$	51.5%
$\frac{9}{16}''$	1.77 to 2.33	2.05	$1\frac{1}{16}''$	$2\frac{3}{16}''$	53.0%
$\frac{5}{8}''$	1.60 to 2.10	1.85	$1\frac{1}{16}''$	$2\frac{7}{16}''$	50.2%

Pitch p of Rivets.—The total strength of a boiler-plate is reduced by the rivet-holes, and the shorter the pitch the weaker the plate, but on the other hand if the pitch is too long the rivet will shear unless it is increased in diameter to correspond in shearing strength to the tensile strength of the net section of plate, but a long pitch and large rivet diameter are also limited by the fact that under high pressures such a joint is hard to make tight.

The mean ratios between the thickness t of plate and the diameter d of the hole given in Table **15 are recommended** as good modern practice.

To find the pitch p in terms of the thickness of the plate t and the diameter d:

$$p = \frac{\pi d^2 f_s}{4 t f_t} + d. \quad \ldots \quad (6)$$

Exercise 30.—Design a single-riveted lap-joint for a boiler 48″ diameter and carrying a steam-pressure of 148 lbs. per square inch, plates to be soft steel of 55,000 lbs. tensile strength per square inch and iron rivets of a shearing strength = 38,000 lbs. per square inch. *Scale 6″ = 1 foot.*

(1) Find thickness t of plate by formula 5, page 132.

(2) Determine diameter d of rivet from the mean ratio in Table 15.

(3) Calculate the pitch p by formula 6, page 136.

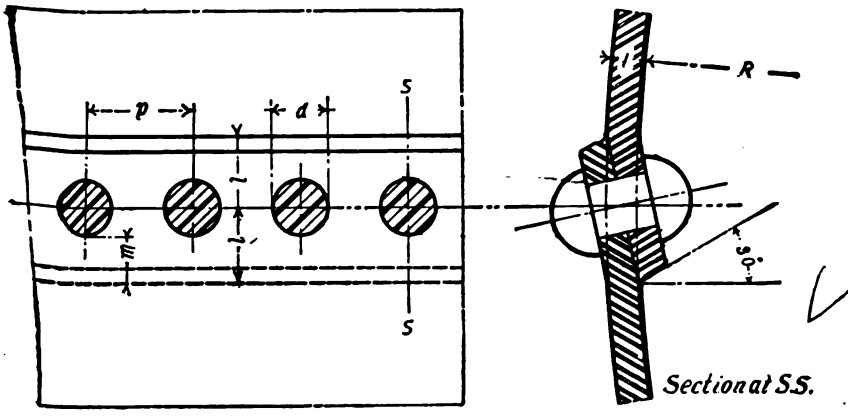


FIG. 97.

Make complete drawings as shown in Fig. 97, giving actual dimensions in place of letters.

Single-riveted lap-joints are commonly used for circumferential seams of steam-boilers.

To determine whether a circumferential seam should be single- or double-riveted let us take the following example:

Diameter of boiler 48″.

Steam-pressure per square inch 148 lbs.

Diameter of rivet = .875″.

Pitch = 2″.

Thickness of plate = .375″.

The total force will be

$$.7854D^2P = 1809.6 \times 148 = 267{,}820.8 \text{ lbs.} \quad . \quad (7)$$

The resistance due to the rivets

$$= \frac{.7854\,d^2 \times n \times f}{F} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

n = the number of rivets in the circumferential seam.

F = the factor of safety = 6.

Therefore, substituting, we have

$$\frac{.601 \times 75 \times 38,000}{6} = 285,475 \text{ lbs.},$$

and, subtracting the force from the resistance, we have a difference of 17,654.2 lbs. in favor of the rivets.

The total resistance of the plate is

$$\frac{(p-d) \times t \times f_t \times n}{F} = \frac{1.125 \times .375 \times 55,000 \times 75}{6} \quad (9)$$

$$= 288,750 \text{ lbs.},$$

and, subtracting the total force, 267,820 lbs., from 288,750, there remains a difference of 20,929 lbs. in favor of the plate, which shows that a single-riveted lap-joint is strong enough for the circumferential seams of a boiler of the above dimensions.

Prof. Lanza referring to the efficiency of riveted joints in his "Applied Mechanics" says:

"A riveted joint of maximum efficiency should fracture the plate along the line of rivets, for it is clear that if failure occurs in any other manner, as by shearing the rivets or tearing out the rivet-holes, there remains an excess of strength along the line of riveting, or, in other words, along the net section of plate—if in a single-riveted joint—which has not been made use of; but when fracture occurs along the net section an excess of strength in other directions is immaterial.

"If the strength per unit of metal of the net section is constant it would be a very simple matter to compute the efficiency of any joint, as it would be merely the ratio of the net to the gross areas of the plate.

"The tenacity of the net section, however, varies and this variation extends over wide limits."

This being so, the pitch in the last example is slightly longer than is necessary.

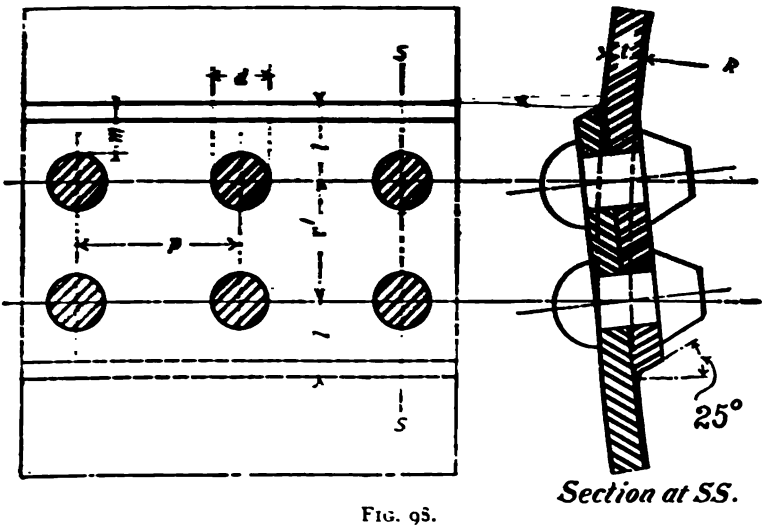
Double-riveted Lap-joints. — The arrangement of the rivets in Fig. 98 is called *chain* riveting and in Fig. 99 *zigzag* riveting. The double-riveted joint is stronger than the single-riveted joint because of the greater net section of *plate* and smaller diameter of rivet-holes. All longitudinal seams in steam-boilers should be at least double-riveted. Steel plates and *iron rivets* are considered the safer practice because of the danger of overheating the *steel* rivets.

Wm. M. Barr in his "Boilers and Furnaces" referring to the heating of steel rivets says: "It is important that steel rivets be uniformly heated throughout, and not the points merely, as is the ordinary method of heating iron rivets; neither should they be heated as highly as iron rivets, and should never exceed a bright cherry-red. Particular attention should be given to the thickness of the fire.

"If excluded from free oxygen steel cannot be burned; if the temperature is high enough it can be melted; but burning is impossible in a *thick* fire with moderate draft."

Chain riveting with rivets of the same pitch has been found by experiment to be stronger than the zigzag riveting. See Barr's "Boilers and Furnaces," page 85, where it states that the lap is wider for chain riveting, "and no doubt the fric-

tion of this wider joint contributes towards the observed crease in strength," but the late D. L. Barnes and others w have tested riveted joints state that the friction between t plates cannot be considered, because long before the ul mate strength of the lap is reached the plates are so far ap that "you can stick a knife-blade between them." The zi zag riveting is preferred in locomotive-boiler seams, becau the joints are tighter under the high pressures carried th they would be with the wider lap of the chain riveting.



Section at SS.

FIG. 98.

Exercise 31.—Make the drawings for a double-riveted lap-joint, chain riveting, like Fig. 98, except that the actual dimensions should be given instead of the letters shown. Steel plates and iron rivets. Thickness of plate $= \frac{5}{8}''$, $p = 3\frac{5}{16}'$, $d = 1\frac{1}{8}''$, $l = 1\frac{1}{2}d$, $r' = 2d + \frac{1}{4}''$, $R = 30''$. *Scale 6" = 1 ft.* Calking need not be shown now.

Calculate the efficiency of this joint in comparison with the strength of the plate.

Taking f_t at 55,000 and f_s at 38,000 as before, the total strength of solid plate is

$$p \times t \times f_t = 3.3125 \times .625 \times 55{,}000 = 110{,}000 \text{ lbs.}$$

The strength of the net section of plate is

$$(p - d)tf_t = (3.3125 - 1.125).625 \times 55{,}000 = 75{,}735.$$

The shearing strength of the rivets $= .7854d^2 \times 38{,}000 \times 2$ (for 2 rivets) $= 75{,}544$, nearly equal to the strength of the net section of the plate. Therefore the efficiency of the joint is equal to

$$E = \frac{75{,}544}{110{,}000} = 69 \text{ per cent nearly.}$$

The following ratios of d to t for double-riveted joints were calculated from the report of a committee on riveted joints to the Am. Ry. M. M. Association in 1895:

TABLE 16.

t	Ratios, Max. and Min.	Ratios, Mean.	d	p	a Area of Rivet.	E
$\frac{3}{8}$" (.375)	2.00 to 2.66	2.33	$\frac{7}{8}$"	$3\frac{1}{16}$"	.6	71.4
$\frac{7}{16}$" (.4375)	1.71 to 2.42	2.06	$1\frac{1}{16}$"	$3\frac{1}{4}$"	.69	69.7
$\frac{1}{2}$" (.5)	1.75 to 2.375	2.063	$1\frac{1}{8}$"	$3\frac{1}{2}$"	.8866	69.6
$\frac{9}{16}$" (.5625)	1.77 to 2.22	1.99	$1\frac{1}{8}$"	$3\frac{9}{16}$"	.994	68.4
$\frac{5}{8}$" (.625)	1.60 to 2.00	1.80	$1\frac{1}{8}$"	$3\frac{5}{8}$"	.994	66.0
$\frac{11}{16}$" (.6875)	1.54 to 1.909	1.72	$1\frac{3}{16}$"	$3\frac{7}{16}$"	1.107	64.7
$\frac{3}{4}$" (.75)	1.416 to 1.75	1.58	$1\frac{3}{16}$"	$3\frac{3}{4}$"	1.107	62.7

To find the *pitch* p for double-riveted lap-joints with steel plates and iron rivets.

$$p = \frac{2 \times a \times f_s}{tf_t} + d = \frac{2 \times a \times 38{,}000}{t \times 55{,}000} + d. \quad (10)$$

To find the distance between the centres of rows of rivets r (Fig. 99).

Prof. Kennedy gives $\frac{2p + d}{3}$ for the diagonal pitch. r may be found graphically or calculated by formula

$$r = \frac{\sqrt{(7p + 2d)(p + 2d)}}{6} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Table 17 gives the distances (r) calculated by this formula for the different sizes of rivets.

Exercise 32.—Make drawings as per Fig. 99 of a double-riveted lap-joint, zigzag riveting. $t = \frac{5}{8}''$, ratio of d to t = 1.80, $R = 30''$. $d = 1\frac{1}{8}t$ in even $\frac{1}{16}''$. *Scale* $6'' = 1$ *foot.*

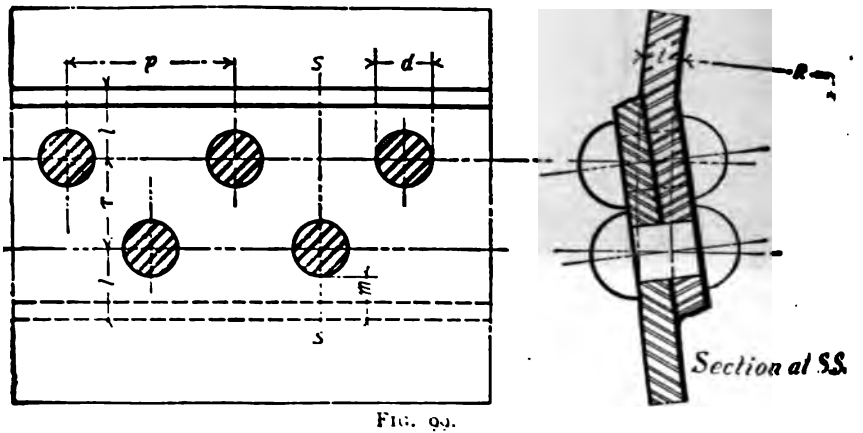


FIG. 99.

Find r by formula 11. Find p by formula 10.

Exercise 33.—Make drawings similar to those in Fig. 100 showing the junction of a double zigzag-riveted longitudinal seam with a single-riveted circumferential seam for a steam-boiler. $t = \frac{9}{16}''$, d calculated from the mean ratio in Table 16, p to be determined from formula 10, p' from Table 14, $R = 29''$, r may be calculated from formula 11. *Scale* $6'' = 1$ *foot.* Actual dimensions to be placed on drawing where letters

show in figure. Steel plates and iron rivets. Finish sheet according to directions given on pages 19 and 20.

Lap-joints with Inside Welt-strip.—This style of riveting, shown in Fig. 101, is used for both single- and double-

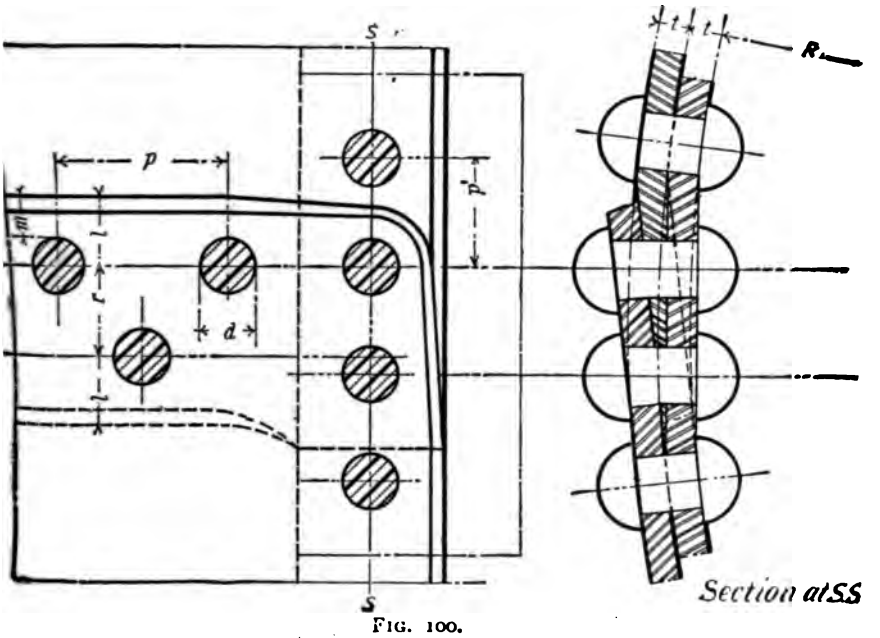


FIG. 100.

riveting and possesses some of the features of the butt- and lap-joint. In the single-riveted joint of this kind the middle row of rivets which rivet the three thicknesses of plate should be spaced according to the rule given for p in the single-riveted lap-joints on page 136 and the spacing of the outer rows $= 2p$.

These joints are better than the simple lap-joint, but are more expensive, and are not any better than the butt-joint (Fig. 102), which is simpler and less expensive.

[illegible]

[illegible] Fig. 101)

[illegible]

[illegible] = [illegible] = [illegible] (12)

[illegible] = [illegible] (13)

[illegible] (14)

[illegible] (16)

$$[illegible] = \frac{[illegible]}{[illegible]} \quad (17)$$

[illegible] — [illegible] Fig. 101. The [illegible]

[illegible] the following table:

[illegible]

[illegible]							Efficiency.
[illegible]						12"	87.0
[illegible]						[illegible]	85.5
[illegible]						[illegible]	85.3
[illegible]						[illegible]	85.0
[illegible]						15"	84.3

The Double-riveted Butt-joint with Inside and Outside Welts.—This style of joint is a very common one for longi-

tudinal seams of steam-boilers with plates $\frac{5}{8}''$ thick and over. As shown by Fig. 102, the boiler-shell is rolled to a perfect cylinder and the two edges of the plate which butt together

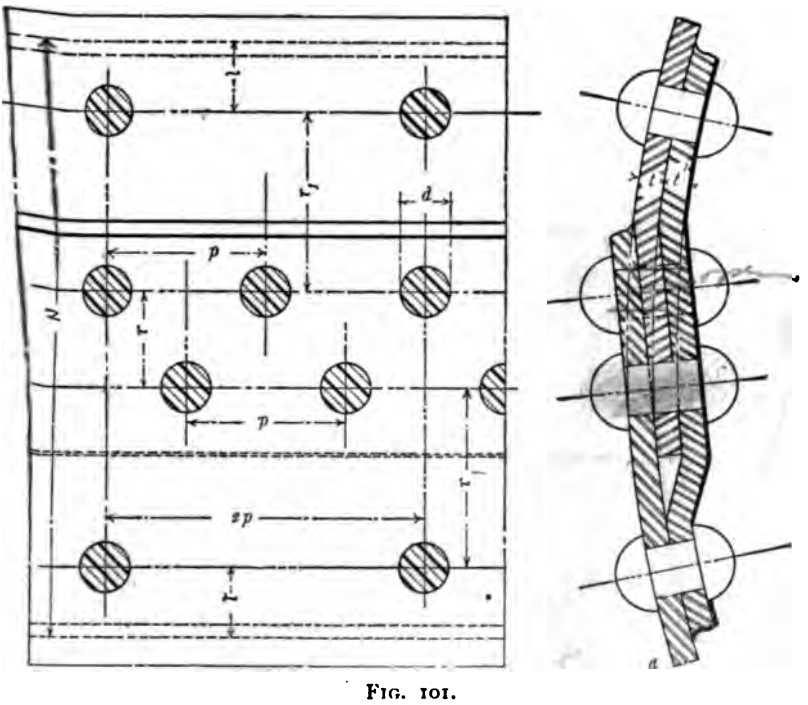


FIG. 101.

are held by two welt-strips riveted to each other and to the ends of the plate.

In a repeating section of the plate = $2p$ there are two rivets in double shear and two half rivets in single shear.

From experiments made by the English Admiralty and others it has been demonstrated that 1 rivet in double shear is equal to 2 rivets in single shear. For convenience we will assume this to be so at present, although it is quite usual for designers of steam-boilers to use a value of from 1.75 to 1.90

for rivets in double shear; and, as the latter values agree more nearly with general practice for butt-joints, it will be necessary for us to modify our proportions in this regard, as will appear later. Therefore to prevent the plate a pulling out from between the welt-strips the resistance to shearing will be

$$5 \times a \times f_s,$$

there being two rivets in double shear and two half rivets in single shear = 5 areas in single shear.

Resistance to tearing the net section of plate at the outer row is

$$(2p - d)tf_t.$$

Resistance to tearing the plate between the inner row of rivets and shearing rivets in outer row is

$$(2p - 2d)t \times f_t + 1af_s.$$

Resistance to crushing the plate in front of 3 rivets is

$$3tdf_c.$$

f_c may be taken at 80,000 lbs. per square inch for iron and 90,000 for steel rivets.

Strength of whole plate equal in width to $2p$ is

$$2p \times t \times f_t.$$

Exercise 35.—Draw *elevation* and *cross-section* of a double-riveted butt-joint with outer and inner welts similar to Fig. 102, given $t = \frac{9}{16}''$, $d = 1.92t$.

If we consider the resistance to tearing equal to the resistance to shearing, then

$$2p = \frac{5 \times a \times f_s}{tf_t} + d, \quad . \quad . \quad . \quad . \quad (18)$$

but this makes the pitch too long, because of the excess of strength in the rivets against shearing. A better proportion

and one that conforms to good practice is

$$2p = \frac{.85(5af_s)}{tf} + d.$$

t_1 and t_2 are usually equal to t, but occasionally t_1 will be

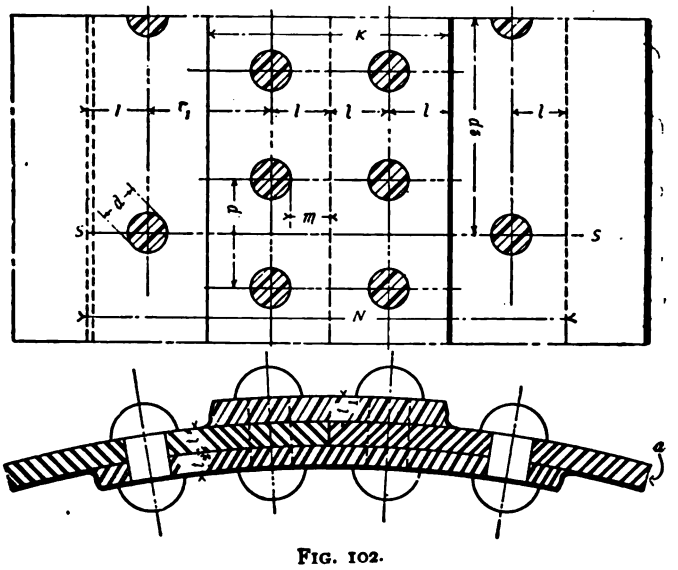


FIG. 102.

found $\frac{1}{16}''$ thicker than t. The Hartford Steam-boiler Inspection & Insurance Company give all welt-strips $\frac{1}{16}''$ less in thickness than t.

For the remaining dimensions see the following table:

TABLE 18.

For double-riveted butt-joints with outer and inner welt-strips.

t	Ratio of d to t. Average.	Diameter of Hole. d	Pitch. p	l	$r_1 = .2l$	K	N
$\frac{3}{8}''$	2.19	$\frac{13}{16}''$	$2\frac{7}{16}''$	$1\frac{9}{16}''$	$2\frac{3}{16}''$	$4\frac{3}{16}''$	$9\frac{1}{8}''$
$\frac{7}{16}''$	1.93	$\frac{7}{8}''$	$2\frac{1}{2}''$	$1\frac{5}{8}''$	$2\frac{3}{16}''$	$5\frac{1}{4}''$	$10\frac{1}{4}''$
$\frac{1}{2}''$	1.92	$1\frac{1}{16}''$	$2\frac{9}{16}''$	$1\frac{11}{16}''$	$2\frac{3}{16}''$	$5\frac{1}{2}''$	$11''$
$\frac{9}{16}''$	1.92	$1''$	$2\frac{9}{16}''$	$1\frac{11}{16}''$	$3''$	$6''$	$12''$
$\frac{5}{8}''$	1.72	$1\frac{1}{16}''$	$2\frac{13}{16}''$	$1\frac{15}{16}''$	$3\frac{1}{16}''$	$6\frac{1}{4}''$	$12\frac{1}{2}''$

Triple-riveted Butt-joint with Outer and inner Welt-strips (Fig. 103).—This joint has three rows of rivets on each side of the butt. One row passes through the boiler-plate and one welt-strip and two rows pass through the sheet and two welts.

The resistance to tearing along line xx is

$$(2p - d)tf_t \quad \ldots \ldots \ldots \quad (19)$$

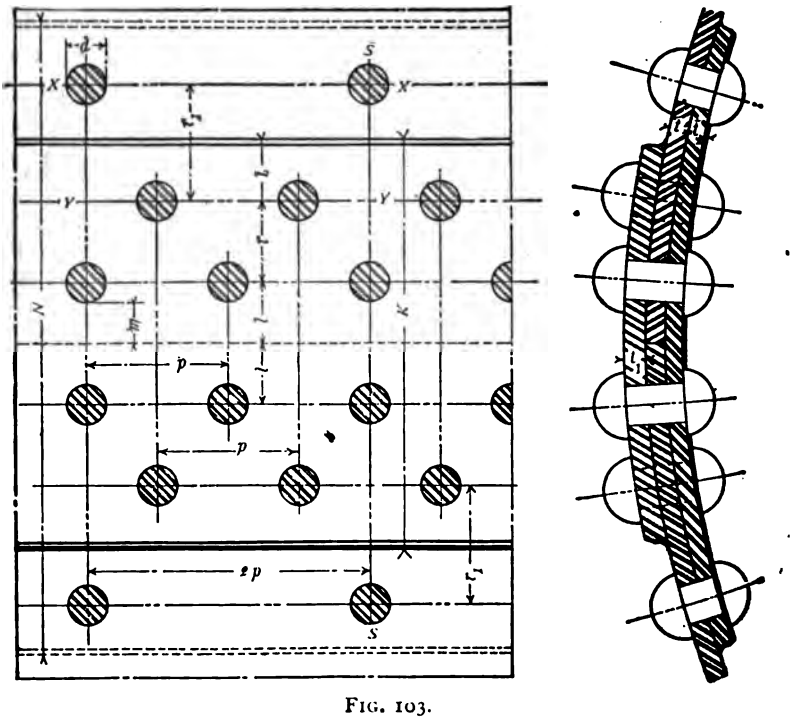


FIG. 103.

The resistance to pulling the plate out **from between the** welt-strips is

$$9 \times a \times f_s \times .85.$$

The resistance to tearing on line *yy* and shearing rivets on *xx* is

$$(2p - 2d)tf_t + 1af_s.$$

A glance at the figure will show that this joint cannot fail along the line *zz*, because there are two rivets in double shear and one rivet in single shear in addition to the net section of plate, which is equal to the net section on *yy*.

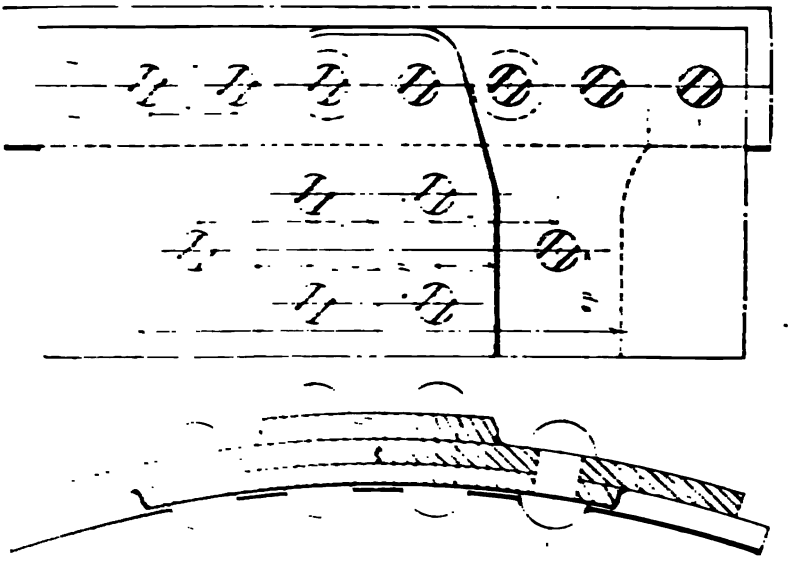
Exercise 36.—Make the drawings for a triple-riveted butt-joint like Fig. 103. Steel plates and iron rivets. $t = \frac{5}{8}''$, $d = 1\frac{1}{4}''$. *Scale* $4'' = 1$ *foot.* The other dimensions may be taken from the following table:

TABLE 19.

t	d	p	m	r	K	N	Efficiency.
In.	In.	In.	In.	In.	In.	In.	Per Cent.
$\frac{3}{8}$	$1\frac{3}{16}$	$2\frac{15}{16}$	$\frac{13}{16}$	$1\frac{11}{16}$	$8\frac{1}{2}$	$13\frac{1}{8}$	86.1
"	$1\frac{7}{8}$	$3\frac{3}{16}$	$\frac{7}{8}$	$1\frac{15}{16}$	$8\frac{7}{8}$	$14\frac{1}{8}$	86.2
"	$1\frac{15}{16}$	$3\frac{3}{8}$	$\frac{15}{16}$	$1\frac{13}{16}$	$9\frac{1}{4}$	$15\frac{1}{8}$	86.1
"	1	$3\frac{3}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{4}$	$16\frac{1}{8}$	86.2
$\frac{7}{16}$	$\frac{7}{8}$	$3\frac{3}{16}$	$1\frac{7}{8}$	$1\frac{13}{16}$	$8\frac{1}{2}$	$14\frac{1}{8}$	86.2
"	$1\frac{15}{16}$	$3\frac{3}{8}$	$\frac{15}{16}$	$1\frac{13}{16}$	$9\frac{1}{4}$	$15\frac{1}{8}$	86.1
"	1	$3\frac{3}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{4}$	$16\frac{1}{8}$	86.2
"	$1\frac{1}{16}$	$3\frac{7}{16}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{8}$	86.2
$\frac{1}{2}$	1	$3\frac{3}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{4}$	$16\frac{1}{8}$	86.2
"	$1\frac{1}{16}$	$3\frac{7}{16}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{8}$	86.2
"	$1\frac{1}{8}$	$4\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{5}{16}$	$11\frac{3}{8}$	$18\frac{1}{8}$	86.1
"	$1\frac{3}{16}$	$4\frac{1}{16}$	$1\frac{3}{16}$	$2\frac{7}{16}$	12	$19\frac{1}{8}$	86.2
$\frac{9}{16}$	$1\frac{3}{16}$	$3\frac{7}{8}$	$1\frac{3}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{8}$	86.2
"	$1\frac{1}{8}$	$4\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{5}{16}$	$11\frac{3}{8}$	$18\frac{1}{8}$	86.1
"	$1\frac{3}{16}$	$4\frac{1}{16}$	$1\frac{3}{16}$	$2\frac{7}{16}$	12	$19\frac{1}{8}$	86.2
"	$1\frac{1}{4}$	$4\frac{1}{16}$	$1\frac{1}{4}$	$2\frac{9}{16}$	$12\frac{1}{4}$	$20\frac{1}{8}$	86.3
$\frac{5}{8}$	$1\frac{3}{16}$	$4\frac{1}{16}$	$1\frac{3}{16}$	$2\frac{3}{16}$	12	$19\frac{1}{8}$	86 2
"	$1\frac{1}{4}$	$4\frac{1}{16}$	$1\frac{1}{4}$	$2\frac{9}{16}$	$12\frac{1}{4}$	$20\frac{1}{8}$	86.3
"	$1\frac{5}{16}$	$4\frac{1}{4}$	$1\frac{5}{16}$	$2\frac{11}{16}$	$13\frac{1}{8}$	$21\frac{1}{8}$	86.1
$\frac{11}{16}$	$1\frac{5}{16}$	$4\frac{1}{4}$	$1\frac{5}{16}$	$2\frac{11}{16}$	$13\frac{1}{8}$	$21\frac{1}{8}$	86.1
"	$1\frac{3}{8}$	5	$1\frac{3}{8}$	$2\frac{13}{16}$	14	$22\frac{1}{8}$	86.2
$\frac{3}{4}$	$1\frac{3}{8}$	5	$1\frac{3}{8}$	$2\frac{13}{16}$	14	$22\frac{1}{8}$	86.2

Calculate efficiency and if possible show where improvement might be made.

Example 3.—[illegible] a longitudinal double-[illegible] and circumferential lap-[illegible] The remaining dimen-[illegible] *Scale* 1 = *1 foot.*



Example 4.—[illegible] the staying for the back-[illegible] motive-boiler as shown

[illegible] shown as the *crown-bar* [illegible] This design is suitable for an [illegible] steam-pressure 180 lbs per [illegible] from the Empire State Ex-[illegible] Buchannan, Supt. of Motive Power of the N. Y. C. & H. R. R. R. shows a cross-section and a partial elevation of the crown-bar which consists of two

wrought-iron plates 5″ deep × ⅜″ thick and welded together at the ends. The fire-box crown-sheet is supported by ⅞″ rivets, which, passing through a washer *b* and between the

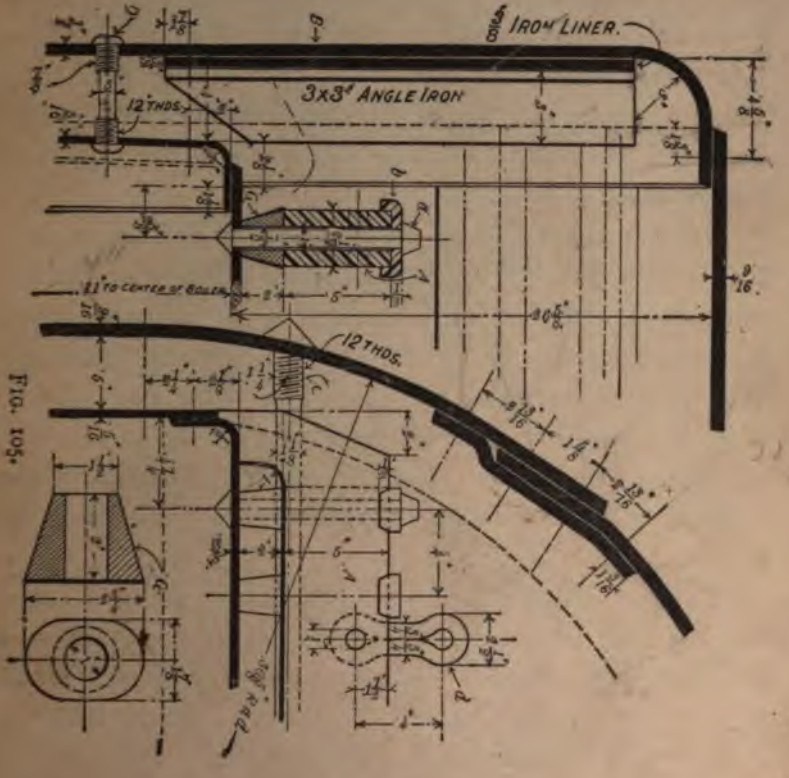


FIG. 105.

plates *A* of the bar and through thimble *G*, is riveted on the under side of the crown-sheet as shown. These rivets are placed from 4″ to 4½″ apart, and as many as the crown-bars will accommodate at these centres, the end bolts being placed about 4″ from the inside of the fire-box side sheets. As seen from the figure, the crown-bars are placed in a transverse position

on the crown-sheet, and as many as the longitudinal length of the sheet will allow, with equal spacing, about $4\frac{1}{2}''$ apart. Should these bars be insufficient to support the crown-sheet against the downward pressure of the steam, which is equal to the area of the crown-sheet × the steam-pressure per square inch, then what remains is held up by *sling*-stays hung from the outer shell and fastened to the crown-bars by links and pins, one link of which is shown at *d* in the transverse cross-section.

The flat upper part of the back-head, which has no stay-bolts passing through it like those which bind the fire-box and outer shell together, as shown at *D*, is stiffened with a liner $\frac{5}{8}''$ thick, the shape of which is shown by dotted lines on the transverse section, and to this liner are riveted as many lengths of $3'' \times 3''$ angle-iron as can be placed on the liner, with a clearance-space of only about $\frac{3}{4}''$ between. To these angle-irons are bolted longitudinal stay-rods $1\frac{1}{8}''$ in diameter similar to that shown in Fig. 106.

To support that curved part of the outside shell just above the fire-box transverse stay-rods *C* are carried between each crown-bar, screwed through the shell on each side, and riveted over on the outside. The body of the rod is $1\frac{1}{8}''$ in diameter and the screwed ends $1\frac{1}{4}''$ diameter.

The fire-box stay-bolts *D* are screwed through both fire-box and outer shell and riveted over outside and inside. It will be seen that while the screwed part of the bolt is $\frac{7}{8}''$ diameter the body is turned down to $\frac{3}{4}''$, which reduces its stiffness and allows it to give somewhat to the unequal expansion of the fire-box and outer shell of the boiler. In certain places the stay-bolts are more liable to break than in others; in such

places hollow stay-bolts are used, so that when broken they may be easily and quickly detected.

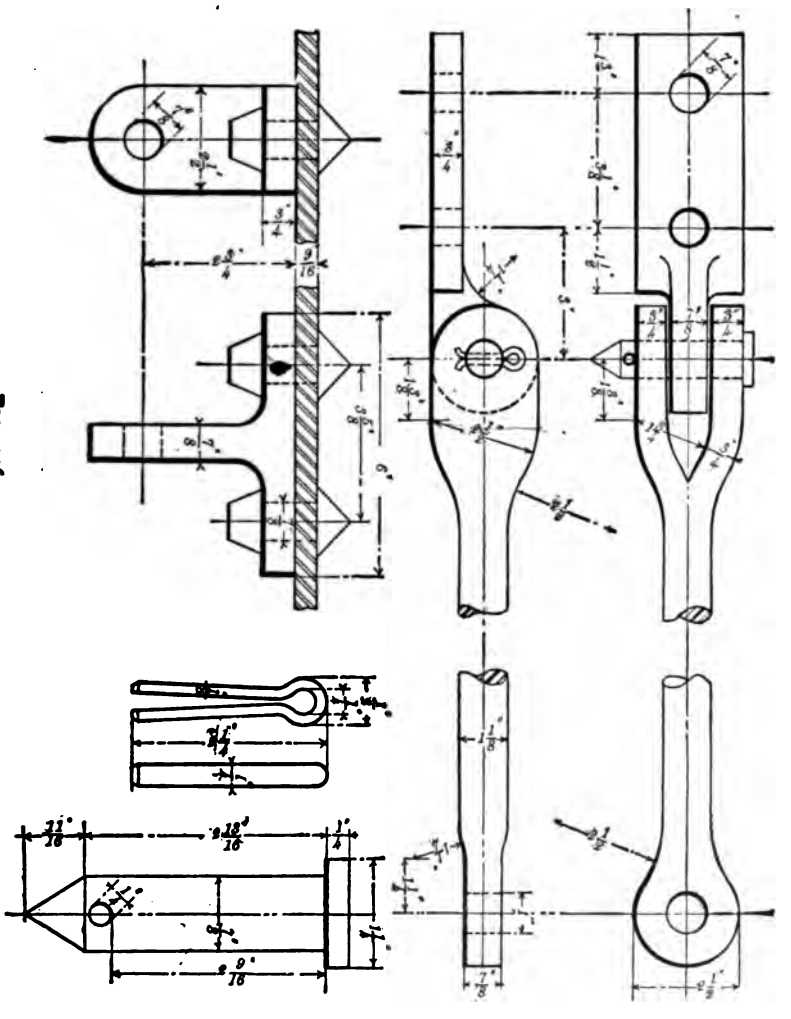


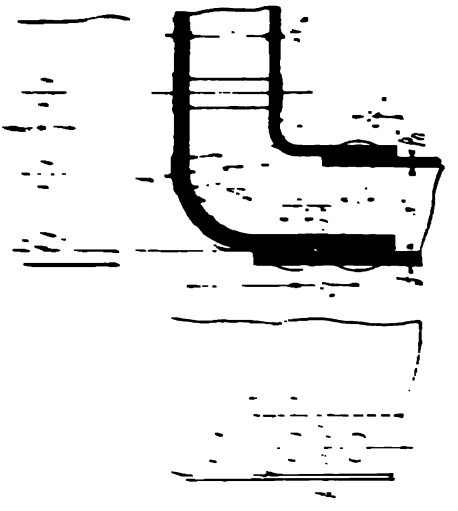
FIG. 106.

Hollow stay-bolts have an $\frac{1}{8}''$ hole drilled completely through from fire-box to the outside of the outer shell, so that

... and water will soon ...

... thimbles is shown ...

... centre line 6′ 6″ ... longitudinal centre line ... the transverse and



... construct the ...

... plates should ... this is a ... without ...

... *longitudinal* ... locomotive-boiler.

Exercise 40.—Figs. 107 and 108 show examples of riveting the corner of a locomotive fire-box ring (sometimes called a

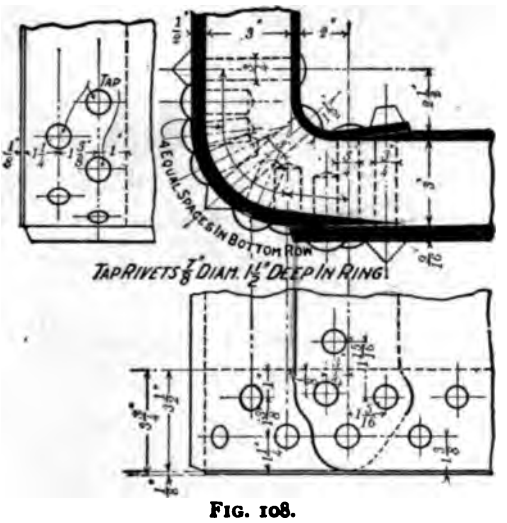


FIG. 108.

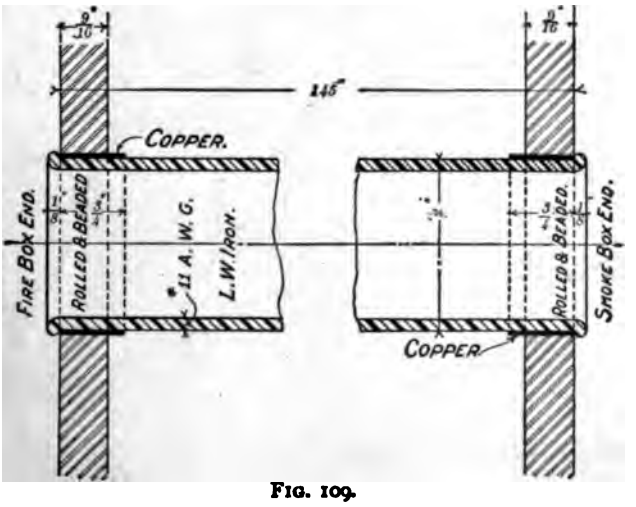


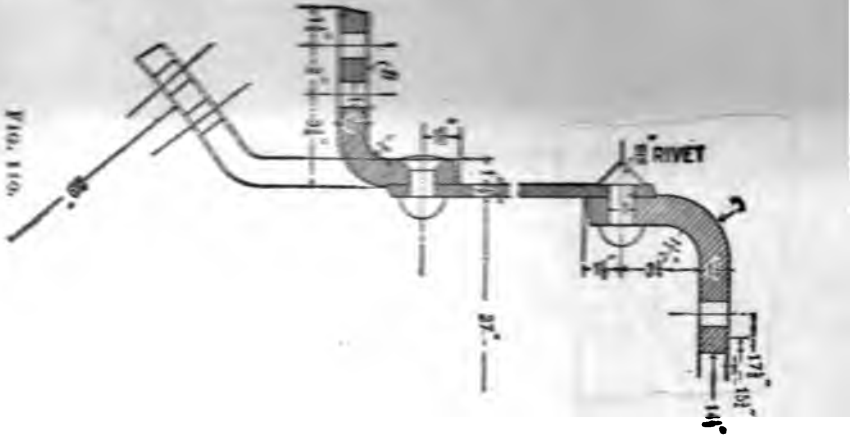
FIG. 109.

mud-ring) to the bottom of the fire-box and outer sh the boiler. Fig. 108 is that of a large boiler 58" dia

and carrying 180 lbs. steam-pressure per square inch, and Fig. 197 is for a smaller boiler of 48" diameter at waist. Make drawings of both figures as shown to a *scale of* $\frac{1}{2}'' = 1$ *foot*.

Exercise 42.—Fig. 199 shows the setting of a tube in the front and back tube-sheets of a fire-tube boiler for a locomotive. Both ends show the tubes swedged, rolled, and beaded, and with copper ferrules between the tubes and the sheets. Make a drawing like that shown by the figure to the *scale of full size*.

Fig. 198 is a section of a locomotive-boiler dome, doming *C* and dome-base *B*. The base and ring are made of soft steel and formed in dies by hydraulic pressure.



Exercise 42.—Make *half sectional elevation, half outside elevation* with *transverse view* and plan. **Also show the curves** of intersection between **the dome-base and boiler.**

CHAPTER IV.

SHAFTING AND SHAFT-COUPLINGS.

UNDER the term *shafting* may be included *line shafting* and *axles*.

Line Shafting.—This name is given to the long line of rotating, cylindrical or square shafting used in workshops and factories for transmitting turning power or *twisting moments* from the prime movers. They are in some ways an extension of the prime mover. Such shafting is subjected to torsional and bending stresses, the latter being due to the pull of belts and the weight of pulleys, gears, levers, etc. "It is usual to make line shafting of uniform diameter throughout, as shown in Fig. 111, enlarged ends being only used occasionally for



FIG. 111.

exceptional purposes. *Steel* of a grade containing .3 to .4% of carbon is now used almost entirely for shafting in preference to *iron* in this country. The commercial lengths of shafting for ordinary diameters, as from 2″ to 3″, run from 16 ft. to 30 ft., the shorter lengths being more convenient for transportation, for replacing pulleys, gears, etc. But the

longer lengths are frequently used when objections do not arise from these considerations." *

Torsional or Twisting Moment.—Figs. 112 and 113 show a lever, a gear wheel and pinion keyed to their respective shafts; R is the radius of the lever and the pitch circle of the gear through which the power P is transmitted. This force P produces a twisting action on the shaft, and the product RP is called the *torsional moment* (T) on the shaft.

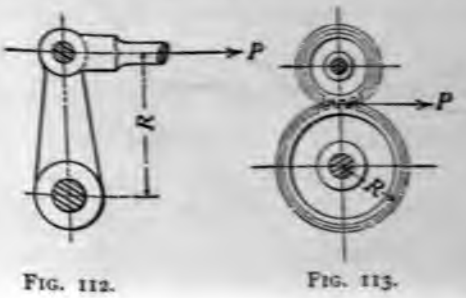


FIG. 112. FIG. 113.

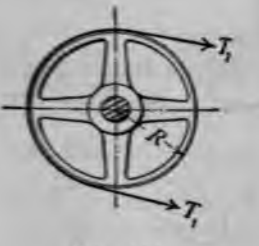


FIG. 114.

So in Fig. 114 P is equal to the tension $T_1 - T_2$, and the radius R multiplied by the force P is again equal to the torsional moment on the shaft. The torsional moment is usually expressed in inch-pounds, i.e., the force P in pounds into the radius R in inches is equal to the torsional moment in inch-pounds.

The moment of resistance to torsion of a cylindrical shaft is equal to the greatest stress multiplied by the modulus of the section.

Let F_s be the greatest shearing stress and Z_t the modulus; then

$$T = F_s Z_t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

* A. & P. Roberts Company.

and $$Z_t = \frac{\pi}{16} d^3 = .19635 d^3;$$

so for cylindrical shafts

$$T = .19635 d^3 f_s, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and for square shafts

$$T = .208 d^3 f_s; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

d = diameter of the cylindrical shaft and length of side of square shaft in inches;

f_s = shearing strength in pounds per square inch;

T = torsional moment in inch-pounds.

To Find the Diameter of a Wrought-iron or Steel Shaft.—If we take the resistance to shearing for iron equal to 40,000 lbs. per square inch and for soft steel at 50,000 lbs., and using a factor of safety of $4\frac{1}{2}$, we have:

For cylindrical iron shafts $T = 1720 d^3$;
For cylindrical steel shafts $T = 2182 d^3$;
For square iron shafts $T = 1849 d^3$;
For square steel shafts $T = 2311 d^3$.

Then

$$d = \begin{cases} \sqrt[3]{\frac{T}{1720}} \text{ for cylindrical iron shafts; } . \quad . \quad (4) \\ \sqrt[3]{\frac{T}{1849}} \text{ for square iron shafts; } . \quad . \quad . \quad (5) \\ \sqrt[3]{\frac{T}{2182}} \text{ for cylindrical steel shafts; } . \quad (6) \\ \sqrt[3]{\frac{T}{2311}} \text{ for square steel shafts. } . \quad . \quad . \quad (7) \end{cases}$$

Example 2.—Let the pitch diameter of a spur gear on an iron shaft be 60", and the total pressure on the teeth at the pitch line 2500 lbs. What would be the diameter of the shaft and the horse-power transmitted by the wheel if running at the rate of 140 revolutions per minute?

From equation (4) we get

$$d = \sqrt[3]{\frac{2500 \times 30}{1720}} = 3\tfrac{1}{4}''$$

for the horse-power transmitted by the wheel.

Let H.P. = horse-power = 33,000 × 12 = 396,000 inch-pounds;

n = number of revolutions per minute;

then

$$\text{H.P.} = \frac{2\pi R \times P \times n}{396{,}000} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$= \frac{6.28 \times 75.000 \times n}{396{,}000} = \frac{6.28 \times 75.000 \times 140}{396{,}000} = 166.5.$$

In terms of the H.P.,

$$T = \frac{63.057 \text{ H.P.}}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and

$$d = \sqrt[3]{\frac{36 \text{ H.P.}}{n}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Besides the twisting stresses on shafts which we have alone taken account of in the above formulæ, there is usually a bending moment to be considered. Let a shaft be subjected

to a torsional moment T and supporting a bending moment B; these two stresses will be equal to a twisting moment

$$T_1 = B + \sqrt{(B^2 + T^2)}. \quad . \quad . \quad . \quad . \quad (11)$$

T_1 is called the *equivalent* twisting moment, and should be used in place of T in figuring the diameter of a shaft subjected to combined torsion and bending. The bending stresses in revolving shafts are continually changing from tension to compression and from compression to tension, so that for combined bending and twisting the factor of safety $4\frac{1}{2}$ given for twisting alone should be increased in the following ratio: When B is more than $.3T$ and not more than $.6T$, the factor of safety should be 5; when over $.6T$ and not more than T, $5\frac{1}{2}$; and when greater than T, 6.

Example 2.—Determine the diameter of a wrought-iron shaft which has to resist a torsional moment of 400,000 inch-pounds and a bending moment of 200,000 inch-pounds. By formula (11) the equivalent twisting moment

$$T_1 = B + \sqrt{B^2 + T^2} = 200{,}000 + \sqrt{200{,}000^2 + 400{,}000^2}$$
$$= 200{,}000 + 447{,}213.5 = 647{,}213.5 \text{ in.-lbs.};$$

and by equation (4)

$$d = \sqrt[3]{\frac{T_1}{1720}} = \sqrt[3]{\frac{647{,}213.5}{1720}} = 7\tfrac{1}{4}''.$$

Example 3.—*When the bending moment exceeds the torsional moment.*—A non-continuous steel shaft has its bearings 8 ft. apart and carries a pulley of 50″ diameter at its centre; the pulley is driven by a 10″ belt, the effective weight and

belt-pull being 500 and 800 lbs. respectively. What should be the diameter of the shaft?

In this case the factor of safety will be 6 and equation (4) becomes

$$d = \sqrt[3]{\frac{T_e}{1634}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$B = \frac{800 + 500 \times 96}{4} = 31,200 \text{ inch-lbs.};$$

$$T = 800 \times 25 = 20,000 \text{ inch-lbs.};$$

$$T_e = 31,200 + \sqrt{31,200^2 + 20,000^2} = 68,230;$$

and

$$d = \sqrt[3]{\frac{68,230}{1634}} = 3\tfrac{1}{2}'' \text{ nearly.}$$

Deflection of Shafting.—A maximum deflection of $\frac{1}{100}$ of an inch per foot of length l for continuous shafting is given as good practice by the Pencoyd Iron Works. The weight of bare shafting $= 2.6d^2 \times l = W$, and for loaded shafts, allowing 40 lbs. per inch of width for the vertical pull of the belts, $W = 13d^2 l$. Then for bending stress alone, taking the modulus of transverse elasticity at 26,000,000, we can derive from authoritative formulæ the maximum length between bearings

$$l = \sqrt[3]{873d^2} \text{ for bare shafts;} \quad . \quad . \quad . \quad . \quad (13)$$

$$l = \sqrt[3]{175d^2} \text{ for loaded shafts.} \quad . \quad . \quad . \quad (14)$$

For line-shafting hangers 8 ft. apart Thurston gives

$$\text{H.P.} = \frac{d^3 n}{90} d = \sqrt[3]{\frac{90\ \text{H.P.}}{n}} \text{ for wrought iron;}$$

$$\text{H.P.} = \frac{d^3 n}{55} d = \sqrt[3]{\frac{55\ \text{H.P.}}{n}} \text{ for cold-rolled iron.}$$

Hollow Shafts.—Weight for weight the hollow shafts are stronger than solid shafts, because the portion of material removed is the least effective in resisting torsion. The resistance to torsion in a solid shaft and a hollow shaft will be equal when the moduli of the sections are equal. Let d be the diameter of a solid shaft, and d_1 and d_2 the internal and external diameters respectively of a hollow shaft. Then

$$d^3 = \frac{d_2^4 - d_1^4}{d_2} \quad \ldots\ldots\ldots \quad (15)$$

A 10″ hollow shaft with hollow 4″ diameter will weigh 16% less than the solid 10″ shaft, but its strength will be only 2.56% less. If the hole were increased to 5″ diameter the weight would be 25% less than that of the solid shaft, and the strength only 4.25% less.

The relation between the weights of solid and hollow shafts is as follows:

Let W = weight of hollow shaft, and W_1 = weight of solid shaft; then

$$\frac{W}{W_1} = \frac{d_2^2 - d_1^2}{d^2}, \quad \ldots\ldots\ldots \quad (16)$$

the weight of the hollow shaft in per cent of the weight of the solid shaft.

An difference in strength is given as follows:

Let strength of hollow shaft, and S_1 = strength of solid shaft; then

$$\frac{S}{S_1} = \frac{d_2^4 - d_1^4}{d_2 d^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

When $d^3 = d_2^3 - d_1^3$ the solid and hollow shafts are of equal weight; and when $d^3 = \frac{d_2^4 - d_1^4}{d}$ the solid and hollow shafts are of equal strength.

A hollow shaft is a solid shaft of the same resistance to torsion, i. not yield to bending as readily; this is an object. some conditions, as in the case of a steamship's propell aft, where it would be an advantage if the shaft would e a little to the straining action of the ship in a storm.

The larger hollow shafts are usually forged hollow, but the smaller shafts are forged or rolled solid and then bored hollow.

SHAFT-COUPLINGS.

We have seen that shafting is made only in short lengths of about 16 ft. to 30 ft. long, so it is necessary in long lines of shafting to couple these lengths together by means of what is known as shaft-couplings. There are three principal divisions of shaft-couplings, viz., *rigid*, *flexible*, and *clutch couplings*. All couplings should be placed close to bearings on the side farthest from the driving-point.

Rigid Couplings.—When two shafts are joined by a coupling that can only be removed by loosening keys or

unscrewing bolts, such a coupling is said to be a rigid or *fast* coupling.

Box or Muff Couplings.—This is the simplest kind of a shaft-coupling (Fig. 115). It is made of *cast iron*. The hole for the shaft is cored small in the casting and afterwards bored out to fit the finished diameter of the shaft. The coupling is secured to the shaft by means of a wrought-iron or steel sunk key about equal in length to the coupling itself, or by two keys each about half as long as the coupling. The latter method is the best, because then it is not so necessary that the keyway in both shafts should be exactly the same depth; moreover, the two keys can be driven tighter and slacked easier than one long key. Half the depth of the keyway is cut in the shaft and half in the coupling. The two shafts butt together at the ends. When two keys are used a clearance space should be left between them when driven home, to insure an equal tightness of both keys, as shown at *S*, Fig. 115.

Exercise 43.—Make a complete working drawing of a muff coupling like Fig. 115 for a $2\frac{1}{2}''$ shaft. *Scale = full size.*

The dimensions may be found from the following proportions:

d = diameter of shaft = $2\frac{1}{2}''$;
t = thickness of metal in coupling = $.4d + \frac{1}{2}''$;
l = length of muff = $2\frac{1}{2}d + 2''$;
$D = d + 2t$.

For proportions and taper of key see page 111. In some positions of the coupling on the shaft the key should have a *gib head*, as shown at *h* in Fig. 117, when it is difficult to

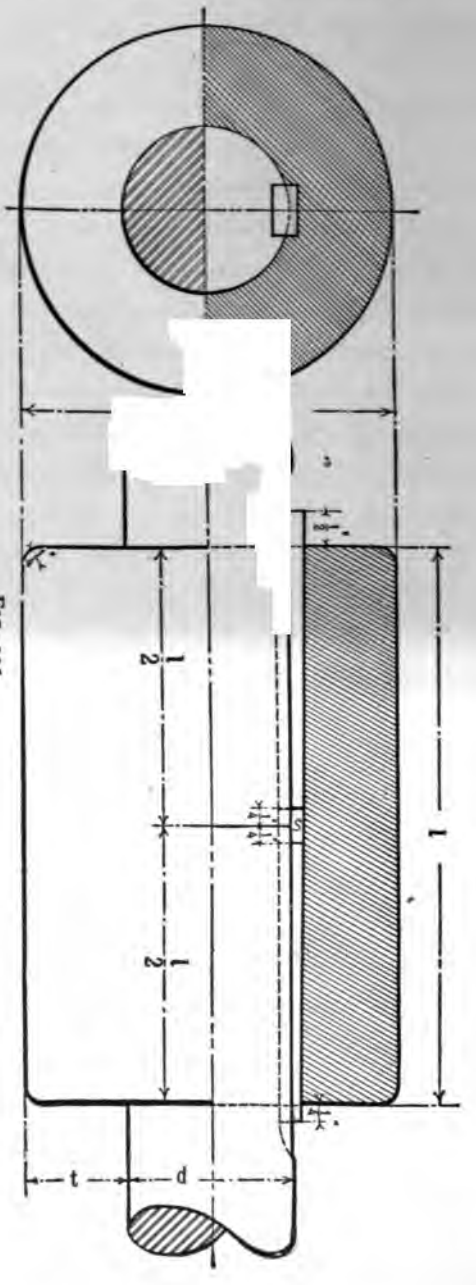


FIG. 115.

obtain access to the small end for the purpose of slacking up. When the coupling is situated close to the bearing it will be necessary to make the length of the keyway to the left of the coupling equal to half that of the corresponding key.

Split-muff Couplings.—Fig. 116 shows a form of coupling divided into two parts and bolted together on the shaft, sometimes called a *compression* coupling. It is keyed to the shaft with a straight parallel key which fits only at its sides. The length of the key may be $\frac{1}{2}''$ longer than the coupling and the keyway the same length as the key.

Exercise 44.—Make working drawings of the split-muff coupling shown in Fig. 116 for a 3″ shaft. *Scale 6″ = 1 foot.*

* In finishing this coupling the inside faces of the two halves are planed and the bolt-holes drilled. Now placing a piece of sheet tin between the halves and bolting them together, the hole is bored for the shaft, making it equal in diameter to the finished shaft. The sheet tin is then removed, and the coupling when bolted on the ends of two shafts clamps them very tightly together. The ease with which the split coupling can be removed and replaced gives it a great advantage over the solid-muff coupling.

Flange or Plate Coupling.—Fig. 117 shows a plate coupling made by the Dodge Mfg. Co. It is made in two parts, of cast iron, and keyed to the two shaft ends, the position of the key in the one shaft being at right angles to the key in the other. The bolts are turned and carefully fitted, and the holes drilled to template so that they can be duplicated if desired. Each coupling should be faced after it has been keyed to its shaft, so as to obtain perfect alignment.

* Check up drawing in Fig. 116 and make corrections.

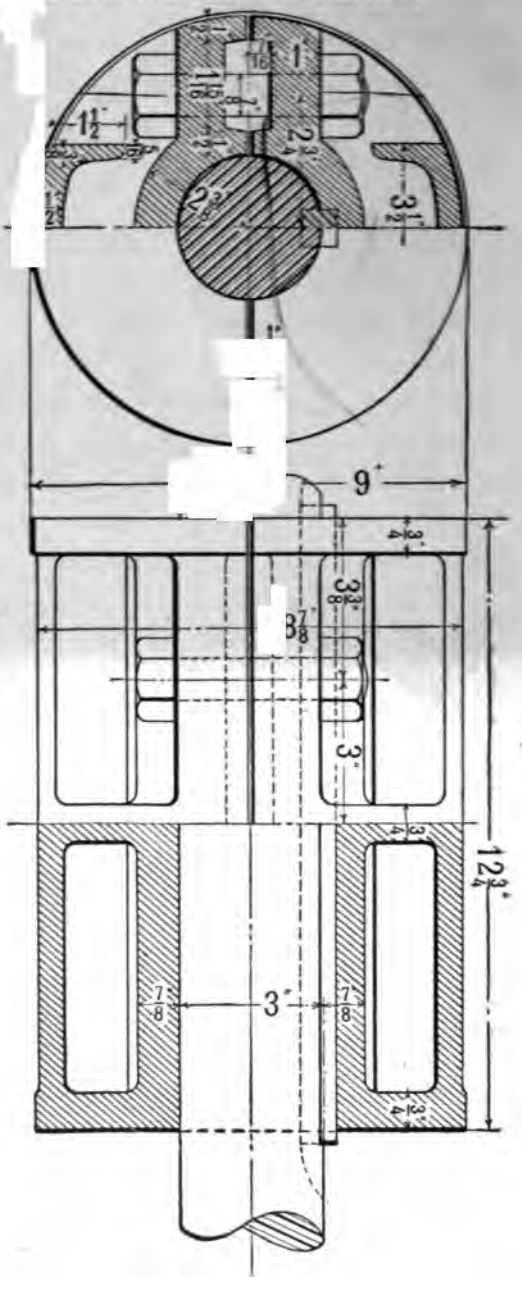


Fig. 116.

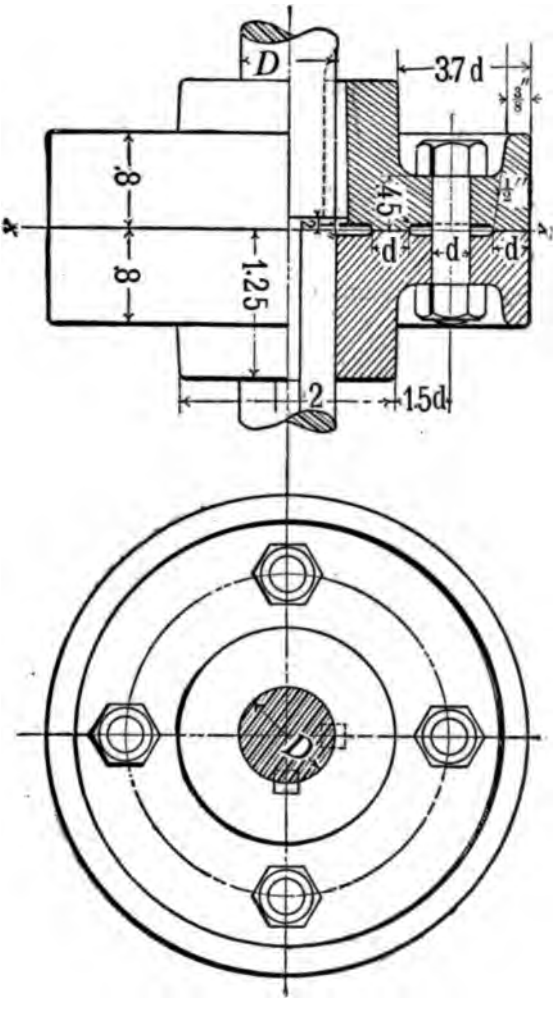


FIG. 117.

Exercise 45.—Make drawings of Fig. 117 as shown, except that the upper half of the end elevation shall be a *sectional* view through *XX*.

Let D = diameter of shaft = 2″;

Unit = $D + \frac{1}{2}''$;

$$n = \text{number of bolts} = 3 + \frac{D}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

Taking the nearest even number,

$$d = \text{diameter of bolt} = \frac{D}{n} + \frac{1}{4}. \quad . \quad . \quad . \quad (19)$$

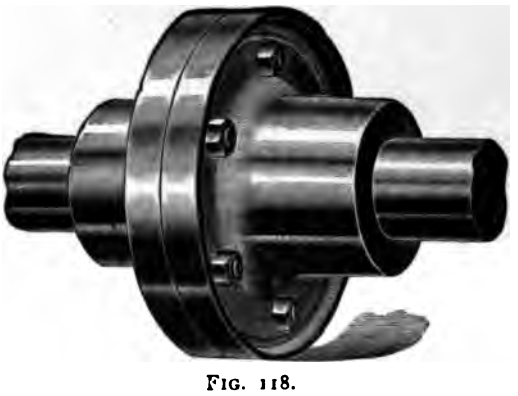


FIG. 118.

The remaining dimensions can be found from the proportions given in the figure in terms of d, the diameter of the bolt.

The taper of the hub may be made equal to $\frac{1}{8}''$ in 12″.

The shaft in this figure is shown sectioned for wrought iron, but in the drawing required it may be sectioned with steel *color*.

Figs. 118 and 119 show plate couplings made by the Hill Clutch Company.

Exercise 46.—Make drawings as required for Exercise 45 of a "Hill" plate coupling for a $4\frac{1}{8}$" shaft. The dimensions for *A* and *B* to be taken from Table 20. The number of

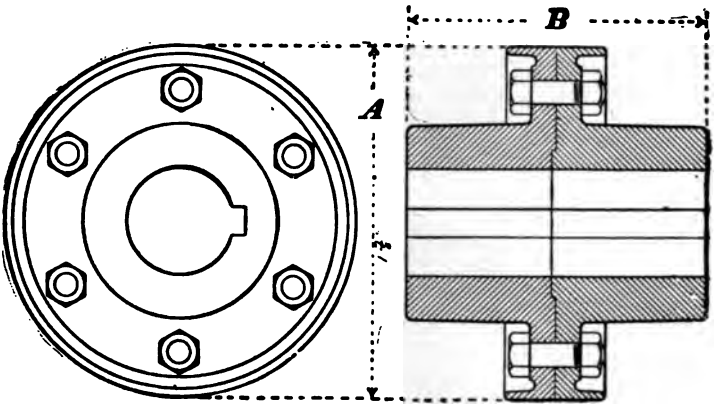


FIG. 119.

TABLE 20.

DIMENSIONS FOR THE "HILL" PLATE COUPLINGS.

Diameter Shaft.	A	B	Diameter Shaft.	A	B	Diameter Shaft.	A	B
$1\frac{3}{16}$	7	6	$2\frac{15}{16}$	$11\frac{1}{2}$	$9\frac{1}{2}$	$5\frac{1}{2}$	$17\frac{1}{2}$	16
$1\frac{7}{16}$	7	6	$3\frac{3}{16}$	$11\frac{1}{2}$	10	6	20	17
$1\frac{11}{16}$	8	6	$3\frac{7}{16}$	$12\frac{1}{2}$	$10\frac{1}{2}$	$6\frac{1}{2}$	21	$18\frac{1}{2}$
$1\frac{15}{16}$	$8\frac{1}{2}$	$6\frac{1}{2}$	$3\frac{11}{16}$	$12\frac{1}{2}$	$10\frac{1}{2}$	7	22	20
$2\frac{3}{16}$	$8\frac{1}{2}$	$6\frac{1}{2}$	$3\frac{15}{16}$	13	$11\frac{1}{2}$	8	24	22
$2\frac{7}{16}$	10	8	$4\frac{7}{16}$	$14\frac{1}{2}$	$12\frac{1}{2}$	9	26	24
$2\frac{11}{16}$	11	$8\frac{1}{2}$	5	$16\frac{1}{2}$	$14\frac{1}{2}$	10	28	26

bolts to be determined from equation (18), and the diameter of the bolts from equation (19) The remaining proportions to be worked out according to the student's judgment.

The Sellers Clamp Coupling (Fig. 120).—This is a special form of a muff coupling which is turned to a cylin-

drical form on the outside, but has a double conical section inside. Two conical sleeves or bushes turned to fit the

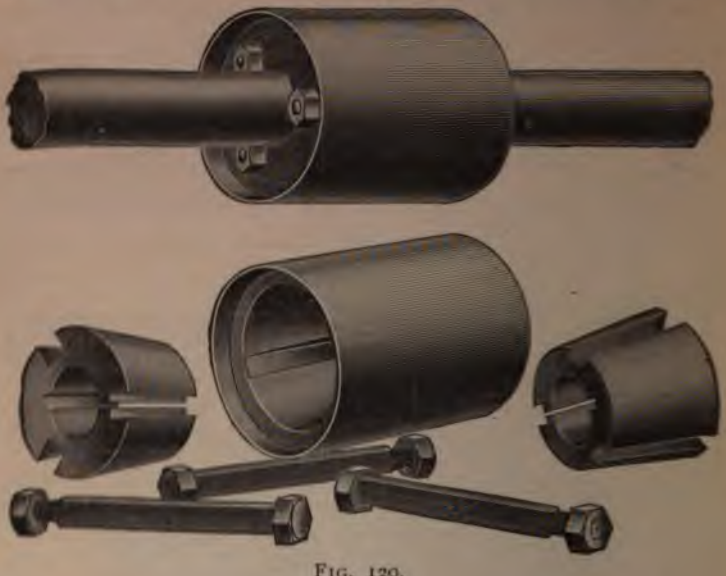


FIG. 120.

inside of the muff and bored out to fit the shafts are pulled together by three bolts. The sleeves are split on one side through one of the bolt-holes, so that the more the bolts are screwed up, the tighter the sleeves clamp the shafts and bind them firmly together. Keys are also used to further prevent slipping.

Exercise 47.—Make the drawings shown in Fig. 121 of a Sellers clamp coupling. *Scale full size.*

The taper of the conical sleeve is $2\frac{3}{4}''$ per foot of length *on the diameter*; e.g., if the sleeve was 6″ long and the large diameter measured 4″, the small diameter would measure $2\frac{3}{8}''$.

For the dimensions of the Sellers clamp coupling for various diameters of shaft, use the following table.

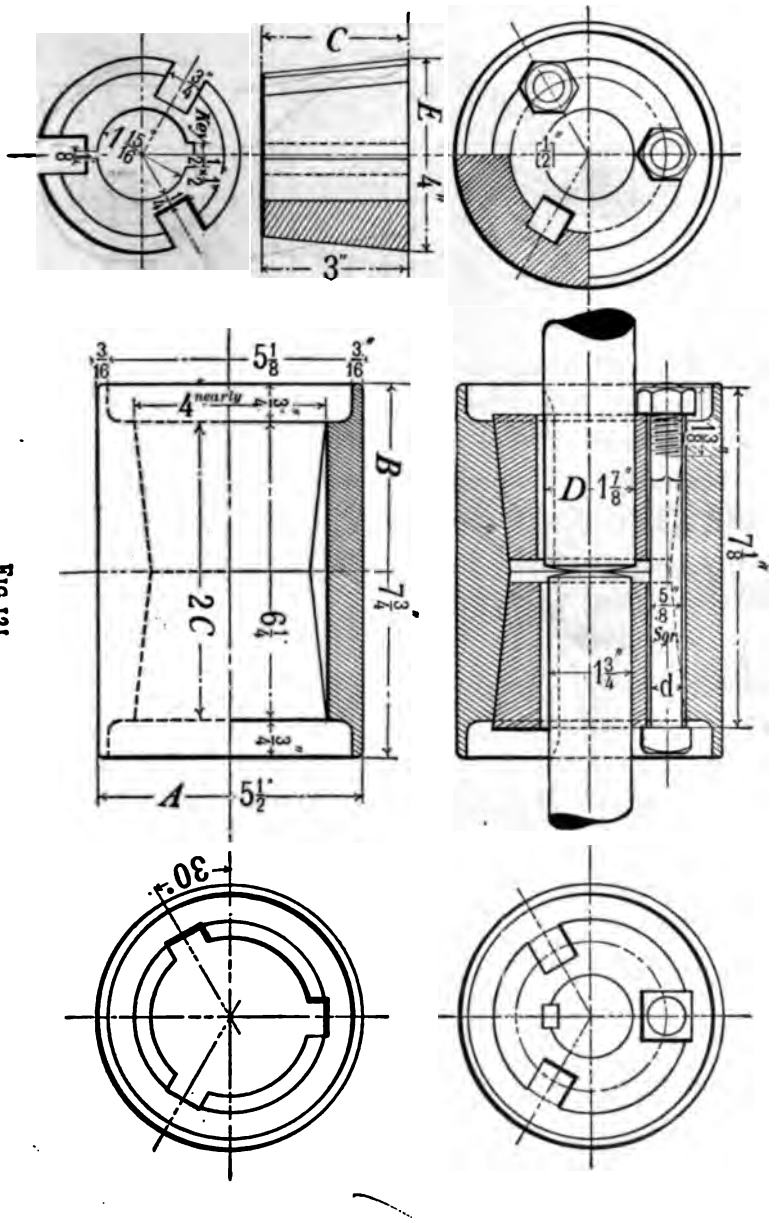

C
E
4"
3"
$\frac{3}{16}$
$5\frac{1}{8}$
$\frac{3}{16}$
4" nearly
B
$7\frac{3}{4}$"
2C
$6\frac{1}{4}$"
$\frac{3}{4}$"
A
$5\frac{1}{2}$"
D $1\frac{7}{8}$"
$\frac{13}{16}$"
$7\frac{1}{8}$"
$\frac{5}{8}$"
Sq.
$1\frac{3}{4}$"
d
30°

Fig 121.

TABLE 21.

SELLERS CLAMP COUPLINGS.

D	*A*	*B*	*C*	*E*	*d*	*D*	*A*	*B*	*C*	*E*	*J*
1½″	4½″	5⅝″	2½″	3⅓	⅞″	3″	8½″	11⅜″	4½″	6¾″	⅞″
1¾	5¼	6⅝	2⅞	4	1	3½	9¾	13¼	5¼	7¾	1
2	6	7½	3	4¾	1⅛	4	11	14⅝	6	8¾	1⅛
2¼	6¾	8¼	3⅜	5	1⅛	5	12⅞	18¼	7½	10¼	1¼
2½	7½	9¼	3¾	5¾	1¼	6	14½	21¼	9	11¼	1¼
2¾	7⅞	10¼	4⅛	6	1⅜						

Frictional Coupling.—Fig. 122 shows three views of Butler's frictional c[illegible]s somewhat like the Sellers coupling, except that i[illegible]r bolts nor keys, the conical bushes being held in p[illegible]ound nuts threaded into the muff. The conica[illegible] at the side, and when they are in position on [illegible] sh[illegible]lit sides are at right angles to each other; this a[illegible]gem[illegible] allows a key-driver to be introduced through one of these [illegible]penings (after the nuts have been removed) to drive out the [illegible]er bush when it is desired to remove the coupling from the shaft. The bushes are guided into position by small dowel-pins which enter short grooves provided for them inside the muff. The ½″ round holes shown in top and bottom at the centre of the muff are used to see when the ends of the shafts come together, for then only will the coupling be in its proper position.

Exercise 48.—Make complete working drawings of the Butler coupling like Fig. 122, except that the shaft shall be of steel and the sectioning shall be appropriately colored instead of hatch-lined. *Scale = full size.*

The threads on the lock-nuts should be that number pe inch used on a pipe whose *outside* diameter is nearest to the

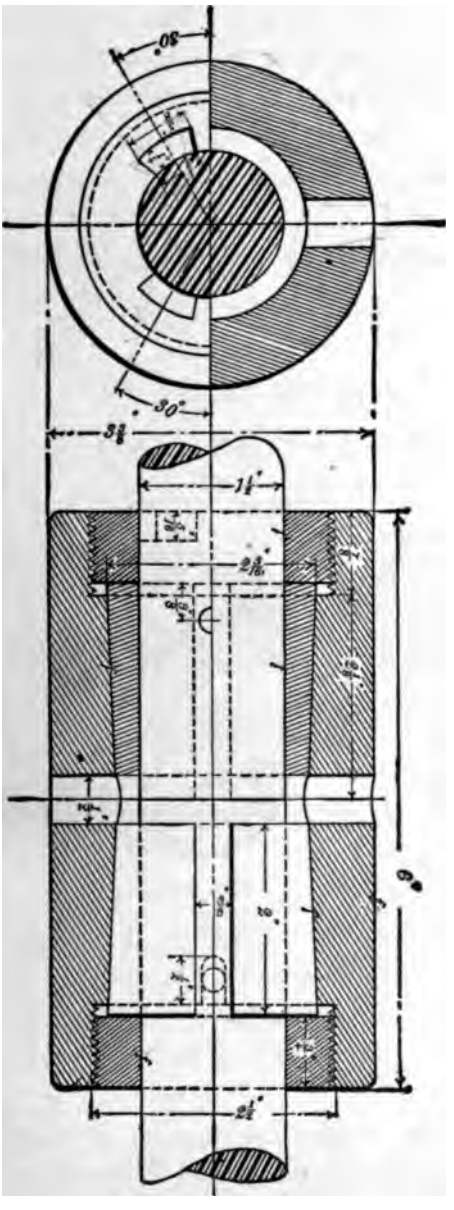


FIG. 134.

outside diameter of the nut. The lock-nuts are screwed into position by means of a spanner wrench having projecting pieces which fit into the recesses shown in end elevation. The taper of the conical bushes may be made $\frac{3}{4}''$ in $12''$ *on the diameter*. The faces marked with small *f* are to be finished.

The principal proportions of this coupling are as follows:

d = diameter of shaft;
D = diameter of muff = $2.25d$;
L = length of muff = $4d$.

Stuart's Clamp Coupling.—This coupling, shown in Fig. 123, differs from the Sellers coupling in having tapered

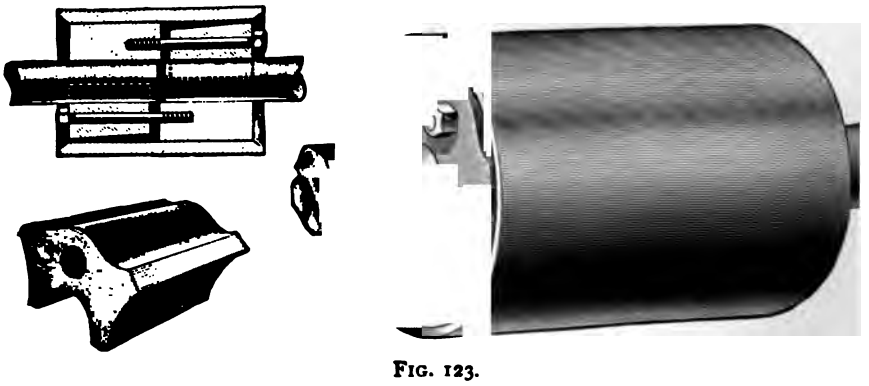


FIG. 123.

wedges instead of conical sleeves; these tapered wedges and opposite halves of each end of the muff are bored to the size of the shaft. Studs and nuts hold the wedges in place, making, on the whole, a cheap and effective coupling without the use of keys.

Exercise 49.—Make drawings of a Stuart's coupling as shown in Fig. 124 for a $1\frac{3}{4}''$ shaft. *Scale = full size.*

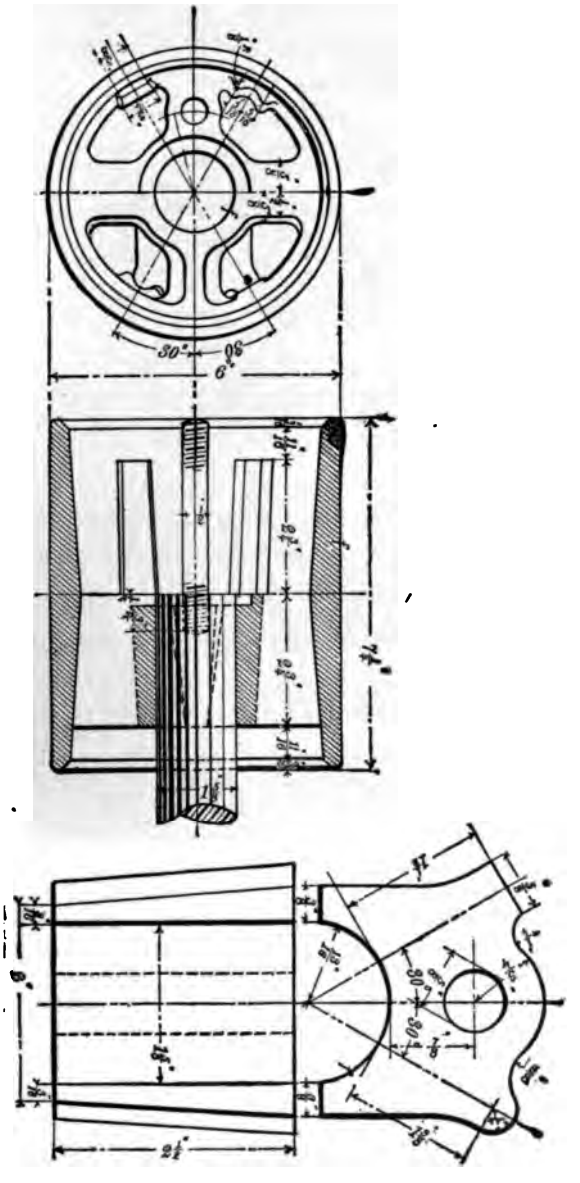


Fig. 124.

The principal dimensions of this coupling for various diameters of shaft are given in the following proportions:

Let d = diameter of shaft;

D = diameter of muff;

L = length of muff.

Then for shafts from 1½" to 2¾" diameter

$$D = 3.25d, \qquad L = 4.25d;$$

for shafts from 2¾" up

$$D = 3d, \qquad L = 4d.$$

Flanged Shaft-coupling.—Fig. 125 shows a propeller-shaft coupling in which the flanges are formed by forging them on the shaft.

Exercise 50.—Make working drawings of the flange coupling shown in Fig. 125. Assume the external diameter of the shaft to be 18" and the internal diameter 10", and take an equivalent solid shaft as the unit for the proportions. *Scale* 2" = 1 *foot.*

Let D_1 and D_2 be the internal and the external diameter, respectively, of the hollow shaft; then from equation (15) we have

$$D = \sqrt{D_2^4 - D_1^4}$$

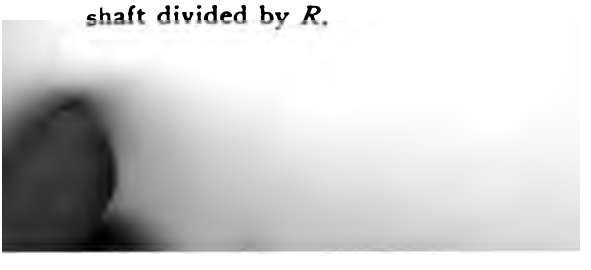
= diameter of an equivalent solid shaft = unit.

d = diameter of bolt;

n = number of bolts = $.25D + 2$;

R = radius of bolt circle = $\dfrac{D + 2.25d}{2}$.

Resistance to shearing of bolts = resistance to torsion of shaft divided by R.



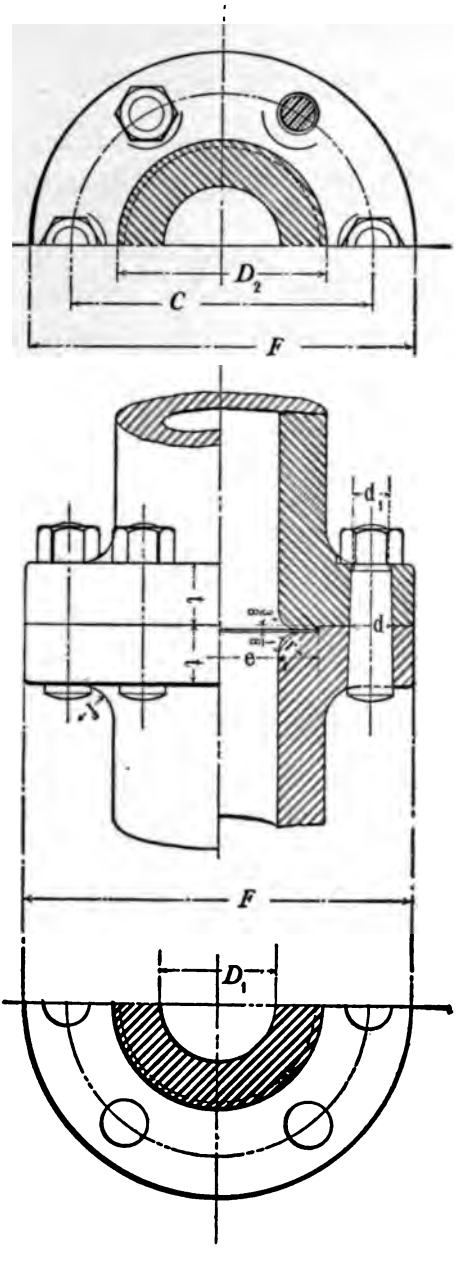
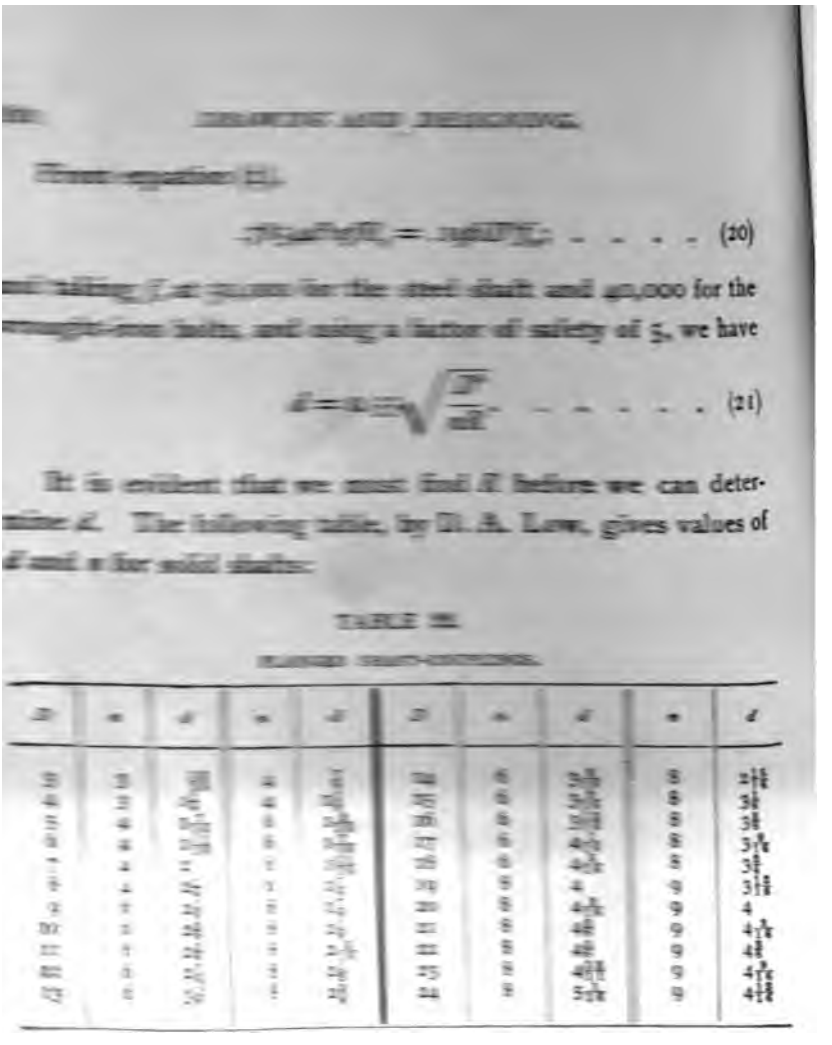


FIG. 135.



d_1 = diameter of screwed part of bolt $= \dfrac{7d + 1''}{9}$;

N = height of nut = $\frac{3}{4}d$ to $\frac{7}{8}d$.

When the bolts are $1\frac{1}{2}''$ diameter or over they are usually tapered, and tapered bolts are often made without heads. For taper of bolt use $\frac{3}{8}''$ in $12''$. (See Exercise 13.)

C = diameter of bolt centre $= D + 2.25d$.

While the shearing resistance of the bolts increases as the diameter C increases with the same diameter of bolt, yet to avoid the unnecessary use of material in the flanges, and secure a maximum of tightness in the coupling, the diameter C should be made as small as it is convenient to make it.

t = thickness of flange = $.3D$;
F = diameter of flange = $D + 3.9d$.
$e = D_1 - 1''$. $f = d_1$.

When cylindrical heads are used in tapered bolts their diameter may be $\frac{1}{4}''$ larger than the largest diameter of the bolt, and the height equal to $\frac{3}{4}d_1$.

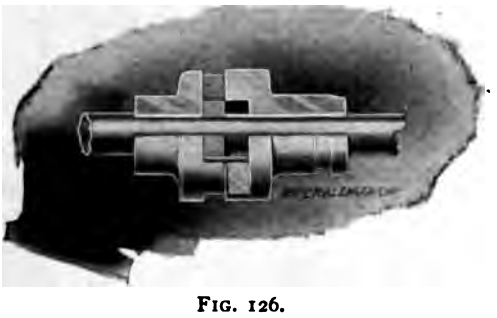


FIG. 126.

Jaw Clutch Coupling.—This coupling, shown in Figs. 126 and 127, is such that one half may be geared with or disgeared from the other half at will, and for slow-moving shafts this arrangement is simple and effective.

Exercise 51.—Make drawings as shown in Fig. 127 for a steel shaft 3″ diameter. The other dimensions may be taken from Table 23.

Spiral-jaw Coupling. This form of coupling, shown in Figs. 128 and 129, has axial engagement and is the commonest

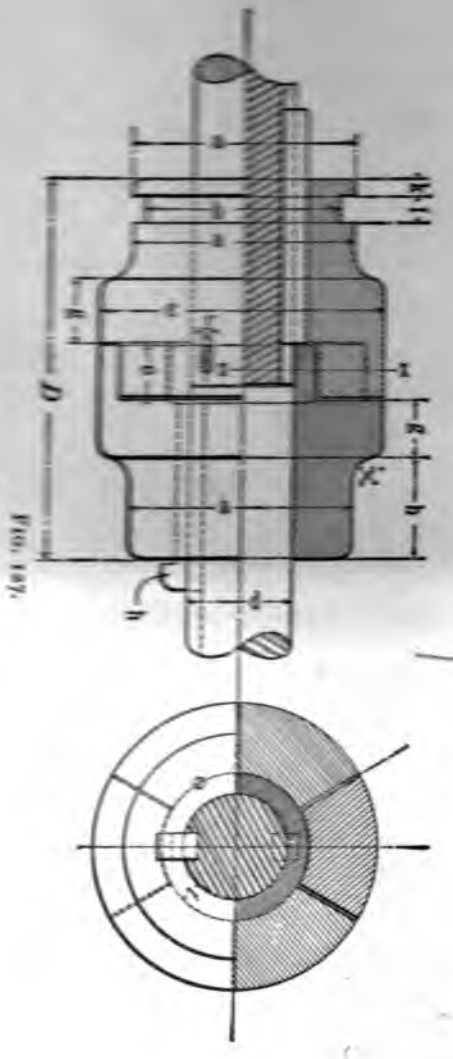


FIG. 197.

TABLE 23.

JAW CLUTCH COUPLINGS.

(Dimensions are in inches.)

a	*b*	*c*	*e*	*D*	*g*	*h*	*t*	*k*	*l*
$2\frac{3}{4}$	$2\frac{1}{4}$	$3\frac{3}{8}$	3/4	$5\frac{1}{8}$	1/2	$1\frac{3}{4}$	1/4	1/4	1/2
$3\frac{1}{4}$	$2\frac{3}{4}$	4	7/8	$6\frac{1}{4}$	5/8	$1\frac{7}{8}$	1/4	3/8	1/2
$3\frac{1}{2}$	3	$4\frac{1}{4}$	15/16	$6\frac{3}{4}$	3/4	$1\frac{7}{8}$	5/16	3/8	1/2
$3\frac{3}{4}$	$3\frac{1}{4}$	$4\frac{3}{8}$	$1\frac{1}{16}$	$7\frac{1}{8}$	7/8	2	5/16	3/8	1/2
4	$3\frac{3}{8}$	$5\frac{1}{4}$	$1\frac{1}{8}$	$7\frac{3}{8}$	1	2	3/8	7/16	1/2
$4\frac{3}{4}$	4	$5\frac{7}{8}$	$1\frac{3}{8}$	$8\frac{3}{8}$	$1\frac{1}{8}$	$2\frac{1}{8}$	3/8	7/16	1/2
$5\frac{1}{4}$	$4\frac{1}{8}$	$6\frac{1}{2}$	$1\frac{3}{8}$	9	$1\frac{1}{4}$	$2\frac{1}{8}$	7/16	7/16	5/8
$5\frac{1}{8}$	$4\frac{1}{8}$	$7\frac{1}{8}$	$1\frac{9}{16}$	$10\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{3}{4}$	1/2	7/16	5/8
6	$5\frac{1}{4}$	$7\frac{3}{8}$	$1\frac{9}{16}$	$10\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{4}$	1/2	9/16	5/8
$6\frac{1}{8}$	$5\frac{3}{8}$	$8\frac{1}{4}$	$1\frac{11}{16}$	$10\frac{7}{8}$	$1\frac{3}{8}$	$2\frac{7}{16}$	9/16	9/16	5/8
$7\frac{1}{8}$	$6\frac{3}{4}$	$9\frac{1}{2}$	$1\frac{15}{16}$	12	$1\frac{3}{4}$	$2\frac{9}{16}$	5/8	5/8	5/8
$8\frac{3}{8}$	$7\frac{3}{8}$	$10\frac{3}{4}$	$2\frac{1}{16}$	13	$1\frac{5}{8}$	$3\frac{1}{4}$	11/16	11/16	3/4
$9\frac{1}{4}$	$8\frac{1}{2}$	$11\frac{7}{8}$	$2\frac{1}{8}$	$13\frac{3}{4}$	2	$3\frac{5}{16}$	13/16	3/4	3/4
$10\frac{1}{2}$	$9\frac{3}{4}$	$13\frac{1}{2}$	$2\frac{1}{2}$	$15\frac{5}{8}$	$2\frac{1}{8}$	$2\frac{5}{8}$	13/16	13/16	3/4
$11\frac{1}{2}$	$10\frac{3}{4}$	$14\frac{3}{4}$	$2\frac{3}{4}$	$16\frac{7}{8}$	$2\frac{1}{4}$	$4\frac{1}{4}$	7/8	13/16	3/4
$12\frac{1}{2}$	$11\frac{3}{4}$	16	3	$18\frac{3}{8}$	$2\frac{3}{8}$	$4\frac{3}{8}$	15/16	15/16	7/8
$13\frac{1}{2}$	13	$17\frac{1}{2}$	$3\frac{1}{4}$	$19\frac{7}{8}$	$2\frac{1}{2}$	5		$1\frac{1}{16}$	1

kind. It is readily thrown in and out of gear by means ever and fork working in a groove shown at the right of



FIG. 128.

igure. This style of clutch is adapted to the transmis- of motion in one direction only.

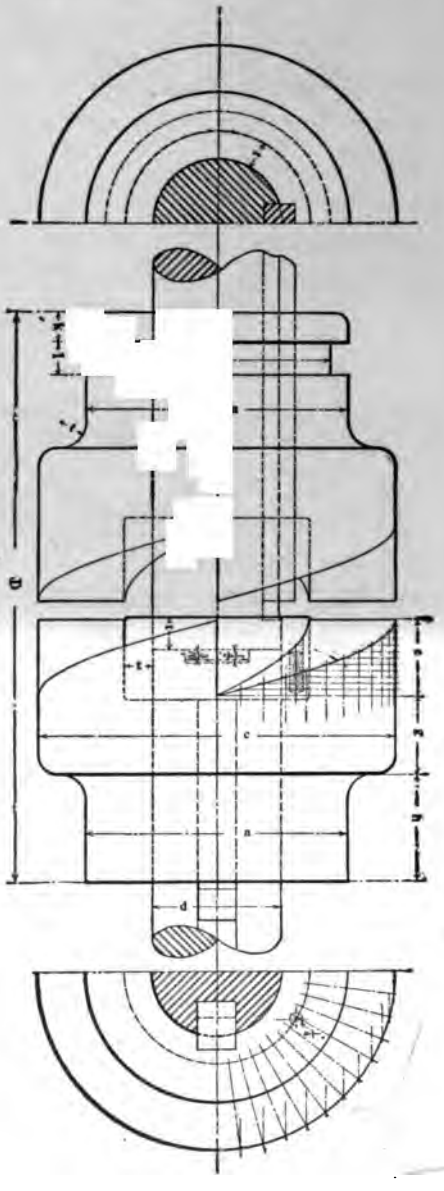


FIG. 129.

Exercise 52.—Make drawings of a spiral-jaw coupling for a $2\frac{1}{2}''$ shaft, as shown in Fig. 129. *Scale* $6'' = 1$ *foot.*

The Universal-joint Coupling.—This most common of flexible couplings is best known as Hooke's coupling. Reuleaux says: "If not the original inventor of the Universal Joint, the Italian Cardan was the first to describe it (1501–1576), and the Englishman Hooke (1635–1702) first applied it for the transmission of rotary motion." The practical value of this form of coupling is that it can be used to connect two shafts whose axes intersect, and the angle between the shafts may be varied during rotation; this latter feature makes it suitable for ship propeller-shafts, to allow for the flexure due to the elasticity of the hull of the vessel.

The coupling shown in Fig. 130 is called a double-joint coupling because of the intermediate piece shown at S, and is such that two shafts in the *same plane* and making equal angles with the intermediate piece (S) will rotate with uniform angular velocity. This coupling is made by the Dodge Mfg. Co., Mishawaka, Indiana.

Exercise 53.—Make complete drawings as shown in Fig. 130. *Scale* $6'' = 1$ *foot.*

Propeller-shaft Coupling.—Fig. 131 shows a propeller-shaft coupling, designed by the Campbell & Zell Co., Baltimore, Md. The material of the coupling and coupling-nut is wrought steel. The design is neat, compact, comparatively inexpensive, and has given good satisfaction.

Exercise 54.—Make drawings of Fig. 131 as shown, except that the diameter of the bolts is to be calculated by equations (20) and (21) (so that the *resistance to shearing* of bolts will be equal to *resistance to torsion* of shaft divided by R),

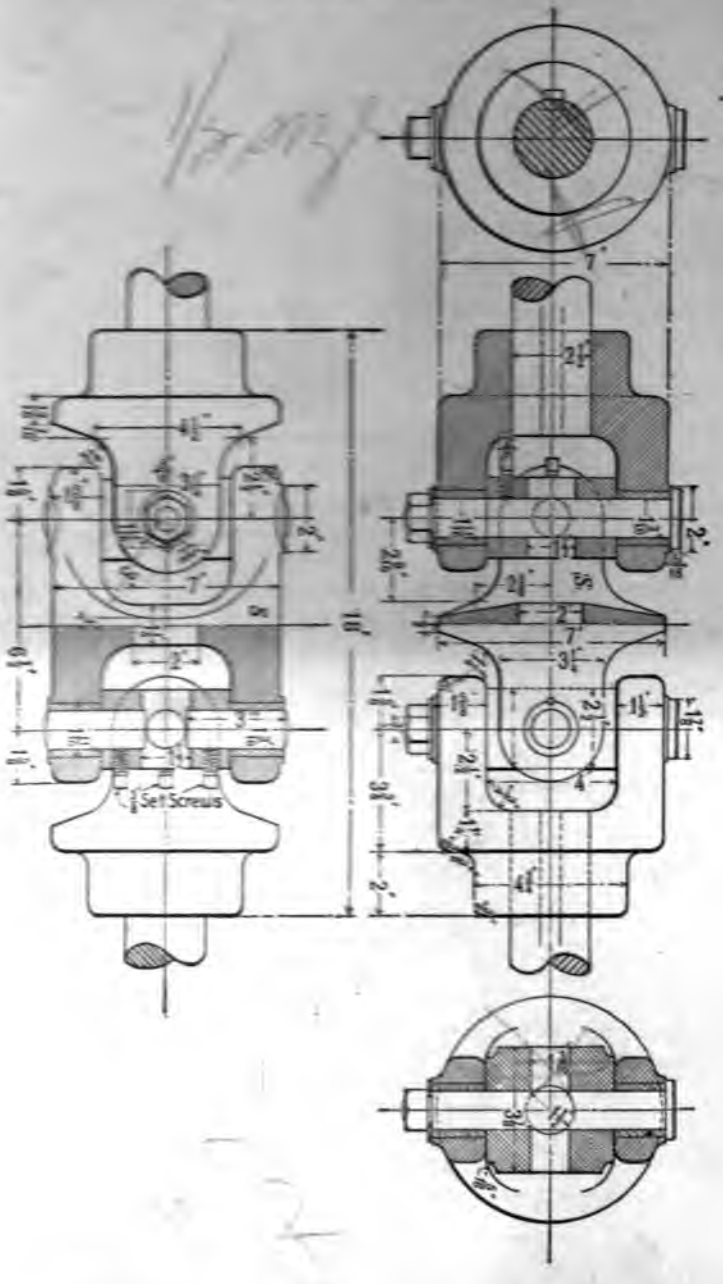


Fig. 130.

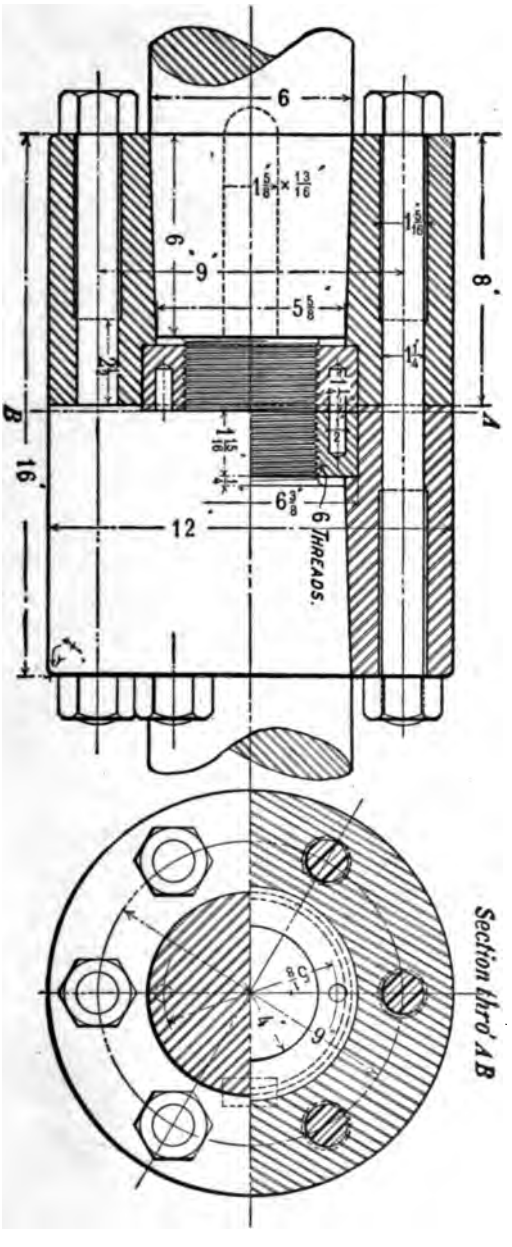


FIG. 131.

and e the diameter of the bolt-centre circle and the out
di f the coupling to suit. *Scale 6″ = 1 foot.*

In ation (20) use D equal to the mean diameter of t
tapered rt of the shaft, and from the result of equati
(21) tak the nearest commercial bolt diameter found
Table 8.

CHAPTER V.

PIPES AND PIPE-COUPLINGS.

Pipes.—Pipes are made of *cast iron*, *wrought iron*, *steel*, *copper*, and *brass*, and used to convey steam, water, or gas. Copper pipes are used most largely in marine work, and brass pipes or tubes are used to some extent in Europe for the fire-tube boilers of locomotives and for other purposes.

Thickness of Pipes to Resist Internal Pressure.

Let D = internal diameter of pipe or mean diameter for very thick pipes;

l = length of pipe in inches, inside of flanges;

P = internal pressure in pounds per square inch;

t = thickness of pipe in inches;

f_t = safe tensile stress in material in pounds per square inch.

Then the total force tending to separate two sections of the shell $= P \times D \times l$, which is resisted by the two thicknesses of the shell $\times$ the length of the pipe $\times$ the pressure per square inch, or $2(ltf_t)$; from this we get

$$t = \frac{PD}{2f_t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This formula gives a thickness somewhat less than is used in practice.

D. A. Low gives

$$t = \frac{PD}{k} + c, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and the values of k and c as follows:

TABLE 24.

	k	c
Cast-iron steam or water pipes	4,000	0.3
Cast-iron steam-cylinders	3,500	0.5
Lap-welded wrought-iron tube	17,000	0.06
Solid drawn steel tubes	40,000	
Copper steam-pipes	7,000	0.1
Lead pipes	450	0.3

For foundry reasons (pipes should never be les
than $\frac{5}{16}''$ thick, and long leng ot less than $\frac{7}{16}''$.

For tables giving the thickness of pipes for various pressures and equivalent heads see Kent's "Mechanical Engineers' Pocket-book," p. 189.

PIPE-COUPLINGS.

Cast-iron Pipe-couplings.—The most common method of connecting cast-iron pipes is by flanges cast on the pipes as shown in Fig. 131.

Exercise 55.—Make drawings of a cast-iron pipe-coupling like Fig. 131. $D = 8''$. Calculate remaining dimensions by the following formulæ. *Scale* $6'' = 1$ *foot*.

$$t = 0.023D + 0.327;$$
$$F = 0.033D + 0.56;$$

$E = 1.125D + 4.25$;
$C = 1.092D + 2.566$;
$d = 0.011D + 0.73$;
n = number of bolts $= 0.78D + 2.56$;
w = weight of pipe per foot $= 0.24D^2 + 3D$;
$W =$ " " flange $= .001D^3 + 0.1D^2 + D + 2$.

This joint has the flanges faced all over, and is used for essures up to 75 lbs. per square inch (170 ft. of head); for

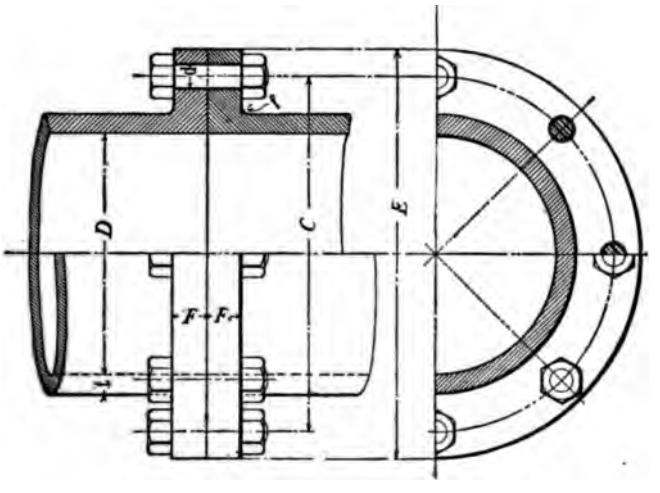


FIG. 131.

gher pressures the joint may be made with a string smeared th red lead between the flanges or a *lead*, *india-rubber*, or *tta-percha* ring.

Exercise 56.—Make drawings of a cast-iron-pipe flange upling, Fig. 132. Inside diameter of pipe to be 9″, other mensions to be taken from Table 25. *Scale 6″ = 1 foot.*

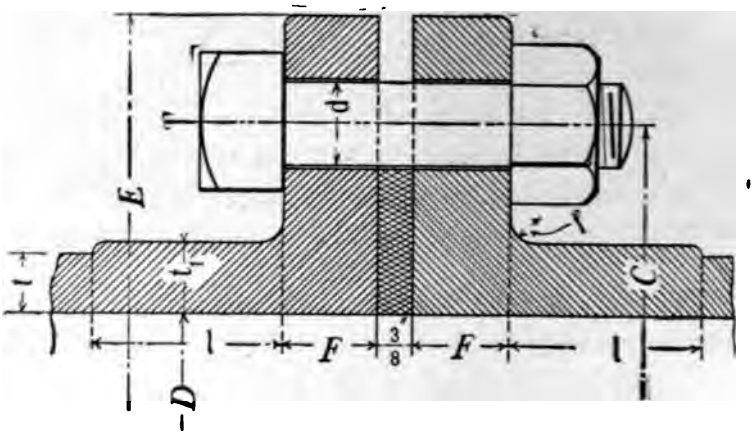


FIG. 132.

$l = 2F. \quad t_1 = t + .2d.$

TABLE 25.

STANDARD CAST-IRON FLANGES.

(Dimensions are in inches.)

D	t	n	C	F	E	d	Stress on Each Bolt per Sq. Inch at Bottom of Thread at 200 Lbs.	f	Stress on Pipe per Sq. Inch at 200 Lbs. Pressure.
2	.409	4	4¾	5/8	6	5/8	825	1/4	460
2½	.429	4	5½	11 16	7	5/8	1050	1/4	550
3	.448	4	6	3/4	7½	5/8	1330	1/4	690
3½	.466	4	6½	13/16	8½	5/8	2530	5/16	700
4	.486	4	7½	15/16	9	3/4	2100	5/16	800
4½	.198	8	7¾	15/16	9½	3/4	1430	5/16	900
5	.525	8	8½	15/16	10	3/4	1630	3/8	1000
6	.563	8	9½	1	11	3/4	2360	3/8	1060
7	.60	8	10¾	1 1/16	12½	3/4	3200	3/8	1120
8	.639	8	11¾	1⅛	13½	3/4	4190	3/8	1180
9	.678	12	13	1⅛	15	3/4	3670	3/8	1310
10	.713	12	14¼	1 3/16	16	7/8	2970	3/8	1330
12	.79	12	16½	1¼	19	7/8	4280	3/8	1470

This table was adopted by a conference of committees of the A.S.M.E. and the Master Steam and Hot Water Fitters Association in July, 1894. Sizes up to 24" diameter are designed for 200 lbs. pressure per square inch or less.

Spigot-and-socket Joint.—This is the usual joint for pipes that have to be embedded in the earth for conveying water or gas. Fig. 143 shows a joint of this kind. About half of the space between the spigot and socket is first filled with rope gasket and into the remaining half is poured molten lead, which when it cools is calked tightly into the socket with a hammer and round-nosed tool.

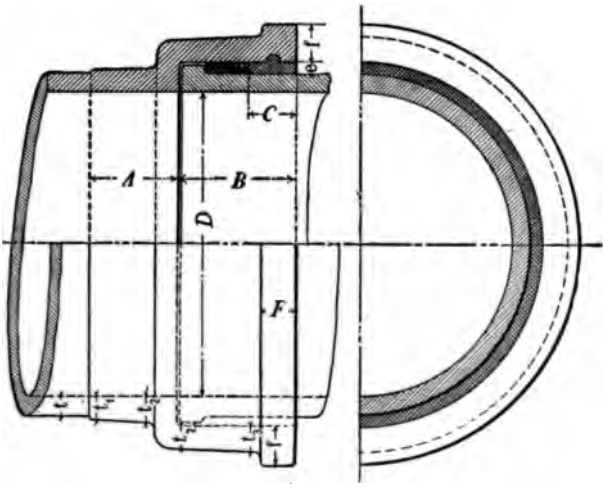


FIG. 133.

Exercise 56 *A*.—Make drawings of a spigot-and-socket coupling for an 8″ cast-iron pipe carrying a pressure of 100 lbs. per square inch (Fig. 133). *Scale* 6″ = 1 *foot*.

Same *elevations* and *sections* as in Ex. 55. Calculate the dimensions from the following proportions:

D = internal diameter of pipe;

t = thickness of pipe $= \frac{PD}{k} + c$ from equation (2);

$$t_1 = t + .2t;$$
$$t_2 = t_1 + \tfrac{1}{8}'';$$
$$t_3 = t_2 + \tfrac{1}{8}'';$$
$$A = .075D + 2\tfrac{1}{4}'';$$
$$B = .1D + 2\tfrac{3}{4}'';$$
$$C = .06D + 1'';$$
$$e = \tfrac{5}{16}'' \text{ to } \tfrac{3}{8}'';$$
$$f = .045D + .8;$$
$$\quad + .7$$

Exercise 57.—Make drawings for the spigot-and-socket cast-iron pipe-co own in Fig. 134. Internal diameter of pipe 10″. ms and *sections* similar to Ex. 55. *Scale* 6″ = 1 *foot.*

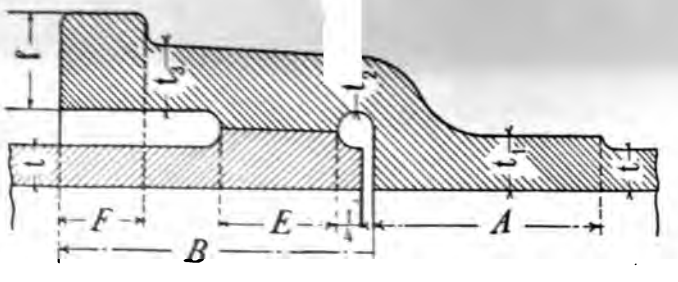


FIG. 134.

The dimensions for this problem are to be calculated from the proportions given for Ex. 56. The turned and fitted part E is made with a taper of $\frac{3}{8}''$ in 12″.

Exercise 58.—Make working drawings of an 8″ cast-iron-pipe flange coupling like Fig. 135. *Elevations* and *sections* as in Ex. 55. *Scale* 6″ = 1 *foot.*

Dimensions to be taken from Table 25.

These pipe-ends and flanges are strengthened with ribs drawn at an angle of 45° with the axis of the pipe, and the

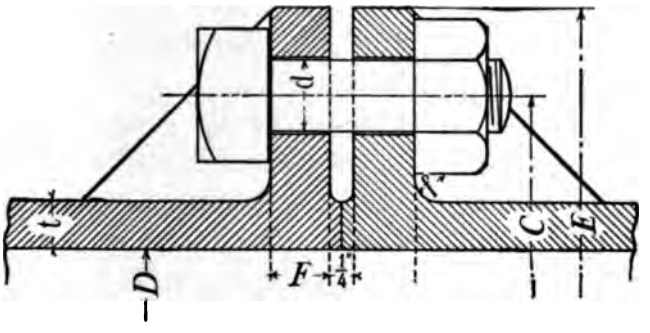


FIG. 135.

joint is made by means of *fitting-strips* cast on the flanges equal in width to the thickness of the pipe. The faces of these strips are finished perfectly square with the axes of the pipes, and before bolting up are smeared with red lead.

Exercise 59.—Make drawings of the *loose* flange coupling for a copper pipe shown in Fig. 136. Inside diameter of pipe 8″. *Scale 6″ = 1 foot.*

This joint is the invention of Mr. R. B. Pope of Dumbar-

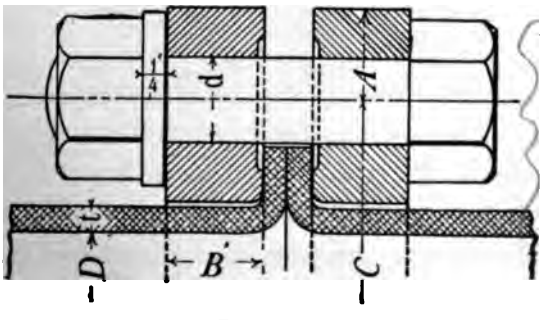


FIG. 136.

tion, Scotland, and is given by Low and Bevis. The flange rings may be made of cast iron, wrought iron, wrought steel, or cast steel; the latter is preferred. It is evident from Fig. 136 that the rings must be placed on the pipes before the ends are flanged.

These joints have been used for steam-, feed-, and exhaust-pipes from $1\frac{1}{2}$" to 36" diameter.

The dimensions may be taken from the following table:

TABLE 26.

POPE'S PIPE COUPLINGS.

(Dimensions are in inches.)

D	A	B	C	d	No. of Bolts.	D	A	B	C	d	No. of Bolts.
$1\frac{1}{2}$	7/8	7/8	4	3/4	5	$4\frac{1}{2}$	15/16	$1\frac{1}{16}$	$7\frac{3}{4}$	7/8	8
2	7/8	15/16	$4\frac{3}{4}$	3/4	5	5	1	$1\frac{1}{8}$	8	1	8
$2\frac{1}{2}$	7/8	15/16	$5\frac{1}{2}$	3/4	6	6	1	$1\frac{3}{16}$	$9\frac{1}{2}$	1	9
3	15/16	1	$5\frac{3}{4}$	7/8	6	7	$1\frac{1}{16}$	$1\frac{1}{4}$	$10\frac{1}{2}$	$1\frac{1}{8}$	9
$3\frac{1}{2}$	15/16	1	$6\frac{1}{2}$	7/8	6	8	$1\frac{1}{16}$	1	$11\frac{1}{2}$	$1\frac{1}{8}$	10
4	15/16	$1\frac{1}{16}$	$6\frac{3}{4}$	7/8	7	9	$1\frac{1}{16}$	$1\frac{1}{16}$	$12\frac{1}{2}$	$1\frac{1}{8}$	10

Wrought-iron and Steel Pipe-couplings. — Fig. 137 shows a very efficient form of joint for wrought-iron pipes. The flanged ends of the pipes are countersunk into the cast flange rings, and the bolt-heads are also countersunk about $\frac{1}{8}$ of an inch. Between the flanged ends of the pipes is placed a ring of lead $\frac{1}{8}$" thick and from $\frac{3}{8}$" to $\frac{5}{8}$" wide.

Exercise 60.—Make drawings of the joint shown in Fig. 137 for a 6" wrought-iron pipe. *Scale full size.*

A should be made equal to $1.25d$. t may be taken from Table 7. Remaining dimensions may be taken from Table 26.

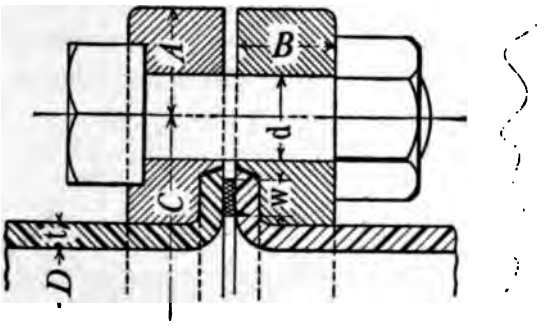


FIG. 137.

The "Converse" joint for wrought-iron and steel pipes is shown at Fig. 138. It is manufactured by the National Tube Works, McKeesport, Pa. This joint consists of a cast-iron sleeve with a space for lead at each end; there are also internal recesses plainly shown in Fig. 138, into which are



FIG. 138.

inserted rivet-heads on the ends of the pipes, and by a turn of the pipes the flanges become locked in position. Molten lead is poured into these recesses around the rivet-heads and tightly calked at the ends of the sleeves, as shown in Fig. 139.

Exercise 61.—Make drawings of the Converse joint for a 7″ wrought-iron pipe, according to the dimensions given in Fig. 139. *Elevations* and *cross-sections* same as in Ex. 55. *Scale 6″ = 1 foot.*

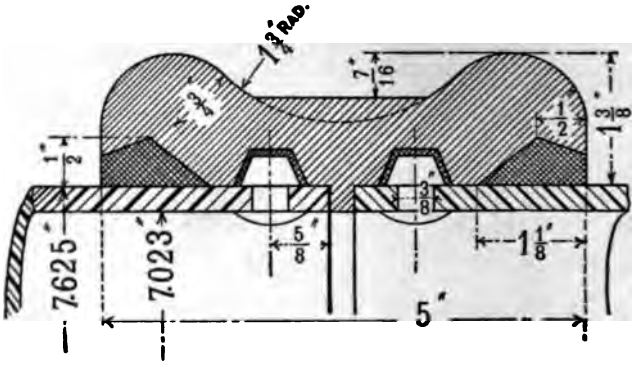


FIG. 139.

Screwed-flange Pipe-coupling. — Fig. 140 shows a wrought-iron pipe-joint made by screwing cast-iron flanges on the ends of the pipes and held together by bolts. It is used by the Philadelphia & Reading Coal and Iron Co. for their steam-pipes. The threads of the screws on the pipes are made according to the Briggs standard. The lugs shown in the figure on the right-hand flange are cast on, and have their inner surfaces finished to fit the cylindrical fitting-piece on the other flange. The ring shown between the flanges is of *gum rubber* and makes the joint steam-tight. The pipes are made in lengths of from 16 to 20 ft.

Exercise 62.—Draw a screwed-flange pipe-coupling like Fig. 140 for an 8″ wrought-iron pipe. *Scale 6″ = 1 foot.*

Dimensions may be taken from the following table:

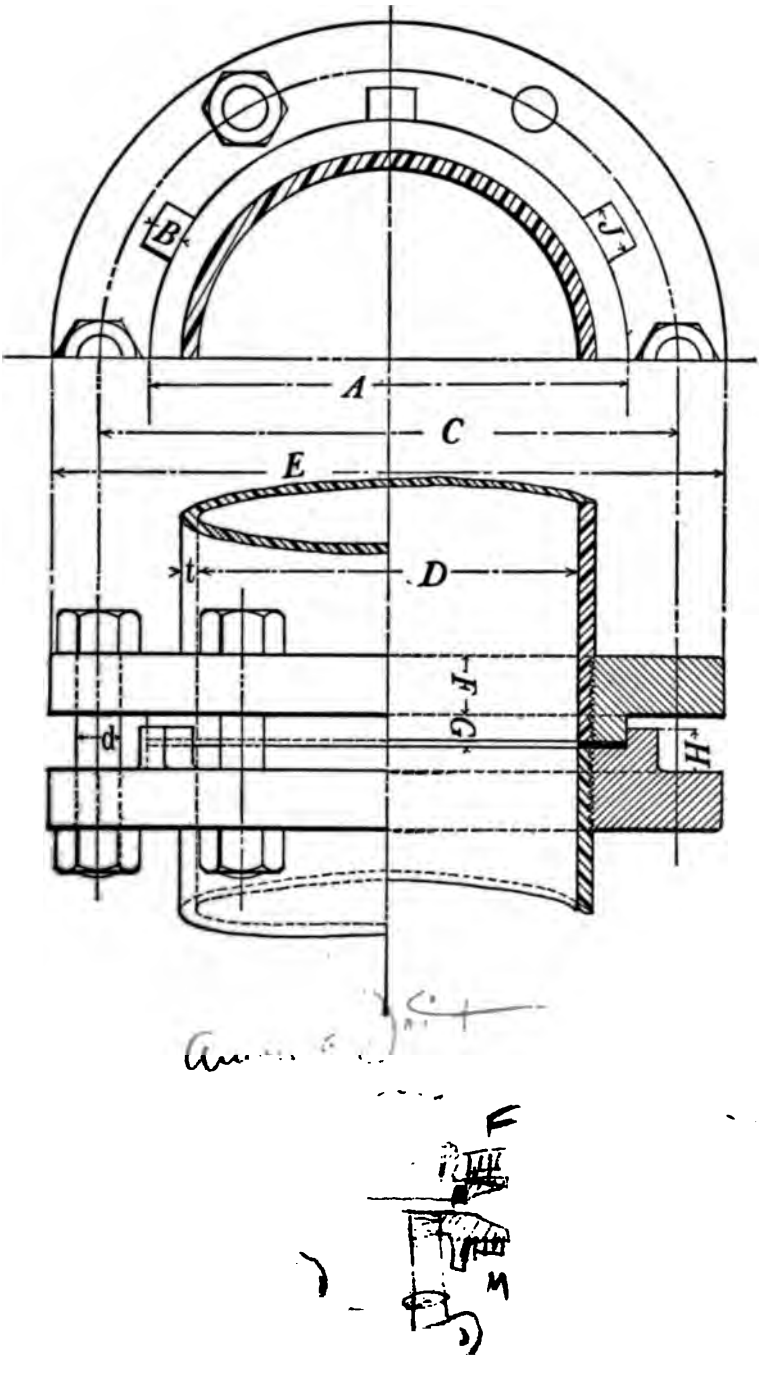

B
J
A
C
E
t
D
F
G
d
H

TABLE 22.

STEAM-PIPE CONNECTIONS OF PELTON & STEARNS COAL AND IRON CO.

(Dimensions are in inches.)

A	F	A	E	C	E	No. of Bolts	a	b	G	H	No. of Lugs.	I
3	3 7/8	5	2 1/2	6	7 1/2	4	3 1/4	3 1/4	1 1/2	7 1/8	4	3/4
3½	4 3/8	5½	2 1/2	6½	8	4	3 1/4	3 1/4	1 1/2	7 1/8	4	3/4
4	4 7/8	6	2 1/2	7	8 1/2	4	3 1/8	3 1/8	1 1/2	7 1/8	4	3/4
5	5 7/8	7	2 1/2	8½	10	4	3 1/8	3 1/8	1 1/2	7 1/8	4	3/4
6	6 7/8	8	2 1/2	10	12	4	3 1/8	3	1 1/2	7 1/8	4	3/4
7	7 7/8	9	2 1/2	11	13	6	3 1/8	3	1 1/2	7 1/8	4	3/4
8	8 7/8	10	3 1/2	12	14	6	3 1/8	2 1/4	2 1/2	7 1/8	6	1
9	9 7/8	11	3 1/2	13	15	6	3	2 1/4	2 1/2	7 1/8	6	1
10	10 7/8	12	3 1/2	14	16 1/2	6	3	2 1/4	2 1/2	7 1/8	6	1

Screwed-socket Coupling.—Fig. 141 shows a screwed-socket coupling for a wrought-iron pipe. The socket is screwed half-way on to the end of one pipe, and the other pipe is then screwed into the remaining half of the socket. When it is not feasible to screw the long lengths of pipe into the projecting ends of the socket, a screw is cut on one length of pipe and the socket is screwed fully on to this length and when the pipes are butted together the socket is screwed back until it is half on each pipe.

For cast-iron, wrought-iron or steel pipe-connections, see samples in drafting-rooms.

Exercise 63.—Make drawings of a wrought-iron socket pipe-coupling 3" nominal diameter, to dimensions given in Fig. 141. *Elevation* and *sections* same as Ex. 55. *Scale full size.*

Locomotive Steam pipe Ball Joint.—This joint (Fig. 142) is made between the steam *branch* pipe (*a*) and the tee-

which conducts the steam from the dome and dry-
the steam-chests of the cylinders on each side of the
The pipes are of cast iron, and the spherical joint-ring

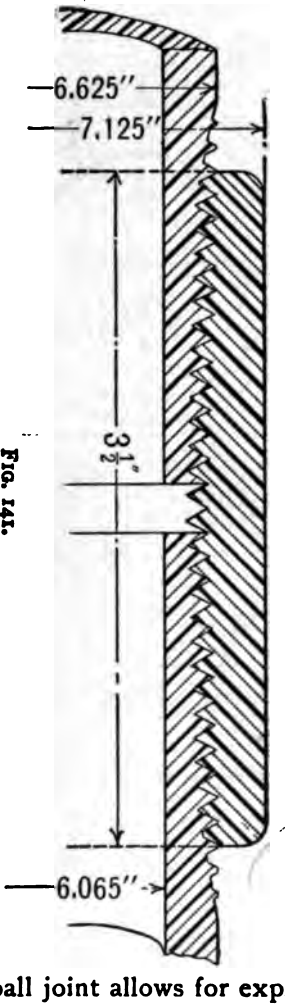


FIG. 141.

ss. The ball joint allows for expansion and contrac-
l for the pipe to be set at various angles with the
icular and horizontal.

Exercise 64.—Make drawings, as shown by Fig. 142, of locomotive steam-pipe ball joint to dimensions given. *Scale* $6'' = 1$ *foot.*

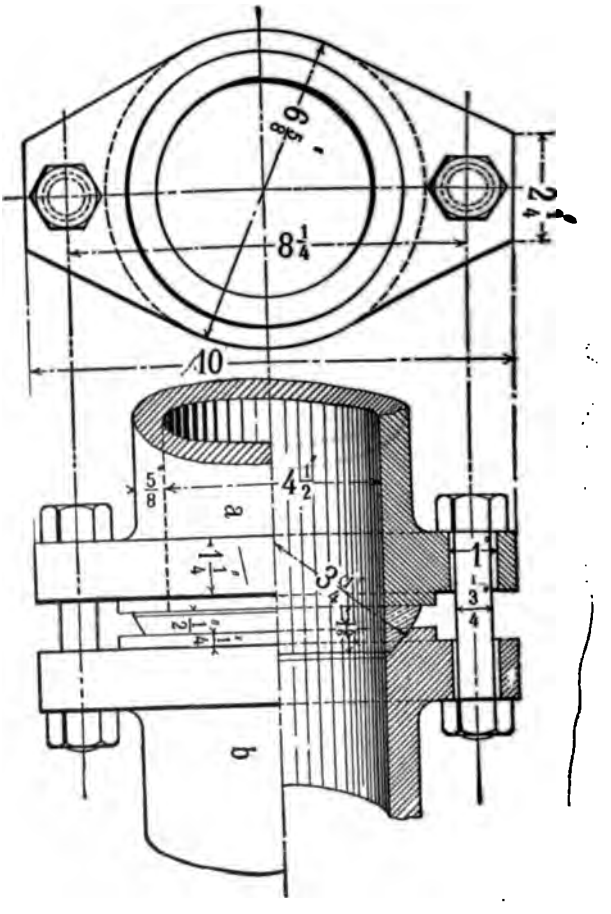


Fig. 142.

Wrought-iron Flange Pipe-coupling.—Fig. 143 shows a pipe-coupling made with angle-iron for a steel pipe. The angle-iron is rolled and welded into rings and riveted to the pipes. These flanges are used for either wrought-iron or

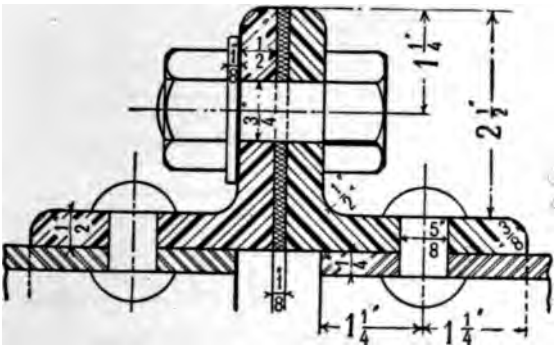


FIG. 143.

steel pipes. The joint is made steam-tight by means of a lead ring inserted between the flanges as shown.

Exercise 65.—Make drawings of a steel pipe with wrought iron flange coupling like Fig. 143. Nominal size of pipe 8″ diameter. *Elevations* and *sections* like Ex. 55. *Scale 6″ = 1 foot.*

Couplings for Brass and Copper Pipes.—The coupling shown in Fig. 144 is used on locomotive-boiler feed-pipes, injector-pipes, etc. The sleeves (*a*) and (*b*) are brazed to the pipes, and a thin copper gasket placed between the ends of the sleeves makes the joint thoroughly tight when screwed up with the fluted nut (*c*).

Exercise 66.—Make drawings, as shown in Fig. 144, of a brass pipe-coupling, outside diameter to be 2¼″. *Scale full size.*

The dimensions may be taken from Table 28.

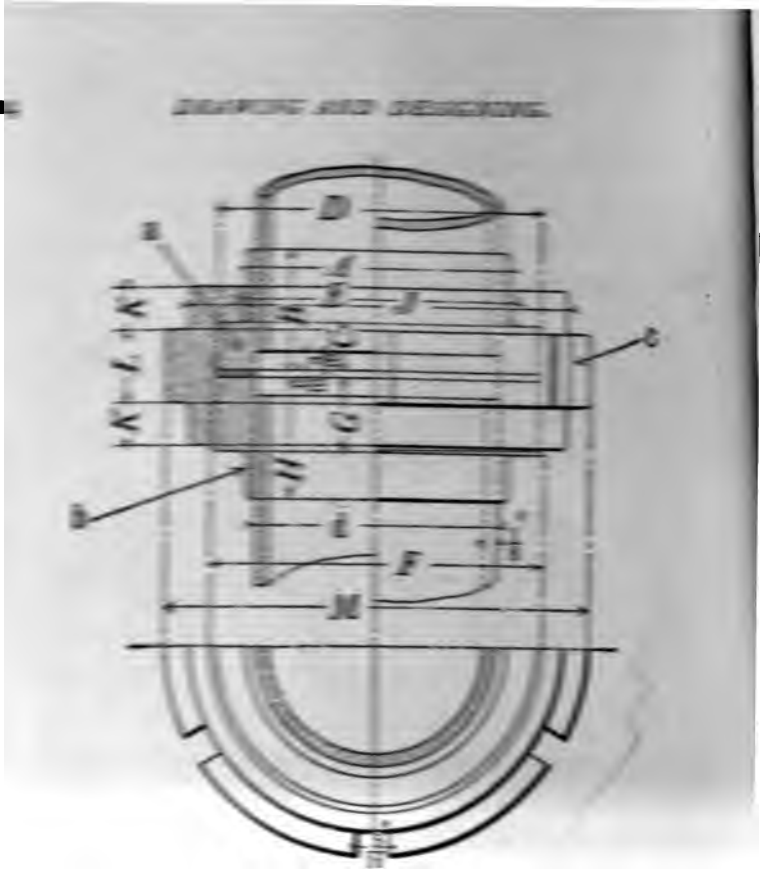


Fig. 244.

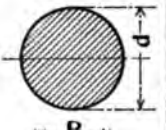
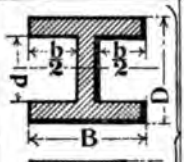
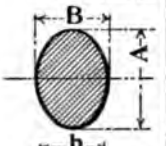
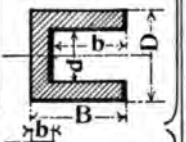
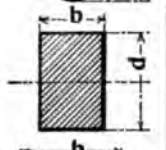
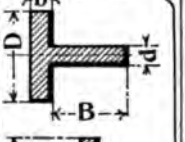
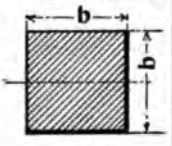
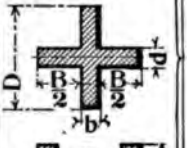
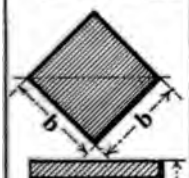
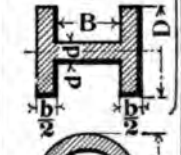
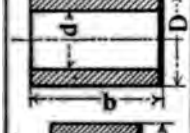
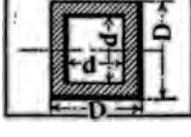
TABLE 24.

CONNECTIONS FOR BRASS, COPPER AND WROUGHT IRON PIPES.

All dimensions are in inches.

d	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*t*
1½	1 7/16	1	5/16	1 1/8	1 7/8	2	3/4	1	2 3/4	3/8	5/8	2 3/4	½
1¼	1 11/16	1	3/8	2 1/16	1 7/8	2 1/4	3/4	1	2 3/4	3/8	5/8	2 1 3/16	
1¼	1 15/16	1	3/8	2 5/16	2	2 1/2	3/4	1	2 3/4	3/8	5/8	3 1/16	
2	2 1/16	1 1/2	3/8	2 1/2	2 1/4	3	7/8	1 1/2	3 3/4	7/16	3/4	3 1/4	
2 1/2	2 7/16	1 1/2	7/16	3 1/16	2 1/2	3 1/2	7/8	1 1/2	3 3/4	7/16	3/4	4 1/4	
2½	2 11/16	1 1/2	1/2	3 5/16	2 3/4	3 3/4	7/8	1 1/2	3 3/4	7/16	3/4	4 3/4	

TABLE 29.

FORM OF SECTION	AREA OF SECTION	MODULUS OF SECTION Z.	FORM OF SECTION	AREA OF SECTION	MODULUS OF SECTION Z.
	$\frac{\pi}{4}d^2 = .7854d^2$	$\frac{\pi}{32}d^3 = .0982d^3$		$DB-d$	$\frac{1}{6}\frac{BD^3-bd^3}{D}$
	$\frac{\pi}{4}BA = .7854AB$	$\frac{\pi}{32}BA^2 = .0982BA^2$			
	bd	$\frac{1}{6}bd^2$			
	b^2	$\frac{1}{6}b^3$		$Db+Bd$	$\frac{1}{6}\frac{bD^3+Bd^3}{D}$
	b^2	$\frac{\sqrt{2}b^3}{12} = .118b^3$			
	$b(D-d)$	$\frac{1}{6}\frac{b}{D}(D^3-d^3)$		$\frac{\pi}{4}(D^2-d^2)$	$\frac{\pi}{32}\left(\frac{D^4-d^4}{D}\right)$
	D^2-d^2	$\frac{1}{6}\frac{D^4-d^4}{D}$		$\frac{\pi}{4}(BD-bd)$	$\frac{\pi}{32}\left(\frac{BD^3-bd^3}{D}\right)$

CHAPTER VI.

BEARINGS, SOLE-PLATES, AND WALL BOX-FRAMES.

ALL pieces employed in the transmission of power, rotating about a geometrical axis, must be supported in such a manner as to allow free rotation. The supports receive the general name of bearings, the various types being designated according to the direction of the pressure acting upon them. When the pressure is perpendicular to the axis of the shaft they are journal-bearings, and when bearings of this type and the framework connected with them are independent parts of a machine, they are indiscriminately called Plummer Blocks, Pillow Blocks, or Pedestals.

When the pressure is parallel to the axis of the shaft and the shaft terminates at the bearing surface, Fig. 164, the bearing is a pivot-bearing. When this type of bearing is employed for supporting the weight of a vertical shaft, it is termed a step- or footstep-bearing. When the pressure is parallel to the axis of the shaft and the shaft is continued through the bearing, the latter is termed a collar-bearing.

When pivot- or collar-bearings are used on horizontal shafts they are called thrust-bearings.

Journals are the parts of the shafts or axles that revolve on the bearings. They are made cylindrical, conical, or spherical, of which the cylindrical is the most common form.

To limit the longitudinal motion of journals the shafts are turned down or have collars forged upon them to form shoulders which come in contact with the faces of the bearings upon which the journals revolve. When practicable the length of journals should be about one per cent greater than that of their bearings.

The Area of a Bearing is the width of the chord of the arc in contact with the journal, multiplied by the length of the bearing. This is sometimes called the projected area, because it is the area of the contact surface projected on to a plane perpendicular to the direction of the pressure. Thus the area of a cylindrical journal-bearing, Fig. 164, is $D \times L$. The area of a pivot-bearing, Fig. 278, is $\frac{\pi D^2}{4}$. The area of a collar-bearing is $\frac{\pi}{4}(D^2 - D_1^2)N$. Where D is the diameter of the shaft D_1 is the outside diameter of collars and N the number of collars.

Solid Journal-bearings.—The simplest form of journal-bearing is made by drilling a hole through the frame of the machine, and to provide sufficient bearing surface the length of the bearing is increased by casting projections, which are termed bosses, upon the frame, as in Fig. 145. In this form of bearing there is no provision for wear, and the shaft can be returned to its initial position only by renewing that part of the frame that carries the shaft, or, when the hole wears oval, reboring the bearing sufficiently to fit it with a cylindrical sleeve or bush, as in Fig. 146. Such a bearing may be provided with a bush or lined with soft metal, and can be restored to its original condition by renewing the bush or lining. The end movement of the shaft may be limited

by making the diameter of one of the journals less than the diameter of the shaft, thereby forming a shoulder which limits the end movement in one direction, and securing a separate collar to the shaft, by means of a set-screw or taper-pin, in such a position as to limit the end movement in the other direction, as shown in Fig. 145. Another method is to make the shaft of uniform section throughout its length, limiting its end motion by means of two separate collars which may be arranged in three different positions.

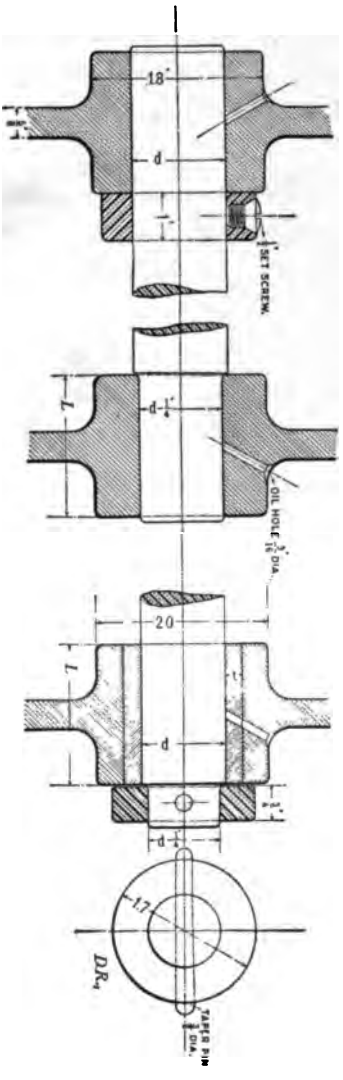


FIG. 145.

FIG. 146.

Exercise 67. — Draw two solid journal-bearings supporting a shaft 2″ in diameter, making the area of the bearing surface 6 square inches, and show an arrangement for limiting the end movement in either direction by means of one loose collar, as shown in Fig. 145. Draw also one bearing $1\frac{1}{4}$″ in diameter with a brass bush or sleeve, as shown in Fig. 146. Parts dimensioned in decimal

Make t equal to $0.1d + \frac{1}{16}$″.

fractions are proportional to d. Complete and fill in the actual dimensions to the nearest sixteenth. *Scale full size.*

As the shafts supported by solid journal-bearings cast with the machine-frame have to pass through one bearing to the other, this form of bearing cannot be used when there are projections on the shaft. A solid bearing can be used, however, for supporting a shaft upon which there are projections, by making the bearings independent parts and securing them to the machine-frame by means of bolts. By this arrangement the shaft is turned down on the ends to form the journals, and one of the bearings is placed on its journal before it is secured to the frame. This form of bearing, Fig. 147, consists of a hollow cylinder cast upon a base through which bolts are passed into the machine-frame or supporting bracket.

Fig. 147 shows a design of a solid journal-bearing used for supporting the valve-gear reversing-shaft of a locomotive. Such a bearing can be used for this purpose because it is subjected to a comparatively light load, while the journal has a slow and intermittent movement. The length and shape of the bearing in this design are determined by local conditions, the bearing being carried forward further on one side of the base than on the other to suit the shaft. The width of the base is determined by the thickness of the frame, and is provided with strips on the under side to facilitate fitting.

Exercise 68.—Draw an elevation and plan of a solid journal-bearing of the form shown in Fig. 147, making $d = 2\frac{1}{2}''$ and $L = 2d$. The parts dimensioned in decimal fractions are proportional to d. *Scale full size.*

Construction.—First draw the centre lines and complete the cylindrical part of the bearing. Make the distance *a* equal to the outside radius of the cylindrical part + *r*, (the

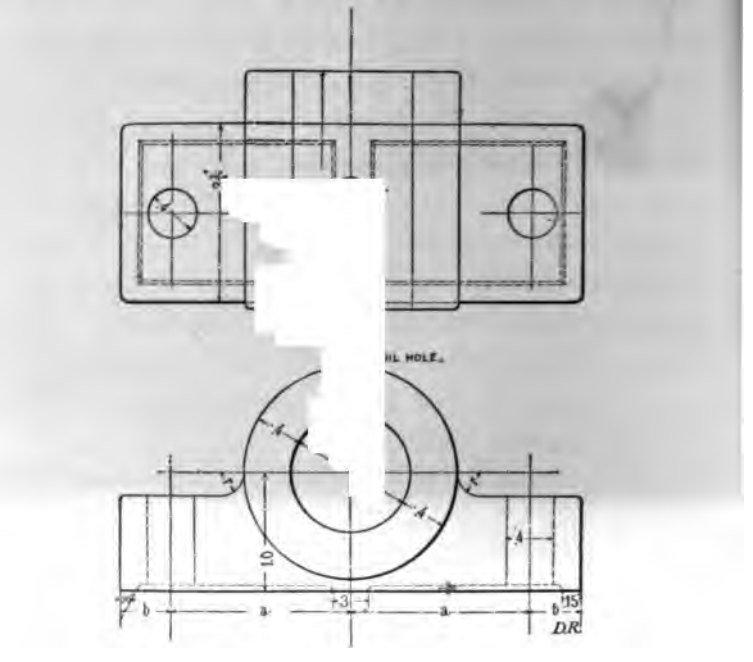


FIG. 147.

radius of the fillet, which we will make equal to, say $\frac{1}{2}''$) + half the distance across the angles of the nut + $\frac{1}{8}''$ for clearance. The distance *b* can be made equal to half the distance across the angles of the nut + $\frac{1}{8}''$.

Divided Bearings —Where the conditions are such that the shaft cannot be placed upon its bearings endwise, the bearings are parted and the parts fastened together by means of bolts or screws. The division is generally made on the line normal to the resultant pressures on the bearing.

In Fig. 148 is shown what is generally termed a two-part bearing. It consists of the block *P*, upon which the journal is supported, and the cap *C*, which is secured to the block by the bolts *CB*. In this design the journal is intended to be lubricated with semi-liquid grease which is passed through the opening *O*. The bearing is lined with Babbitt metal, $.c8D + \frac{1}{16}''$ thick. The holes through which the holding-down bolts pass are made oblong to horizontally adjust the pedestal.

Wall Box-frames are built into the wall for the purpose of supporting a bearing for shafting which passes from one room or building to another. Fig. 149 shows a wall box-frame with an arched top to support the wall above it. On



FIG. 149.

the sides are cast projecting webs *W* which fit into the wall to keep the frame from moving endwise. The upper side of the base is provided with raised machined strips *FS* upon which the pedestal rests, as shown in Fig. 150, and at each end of this surface are projections *S*, on the sides of the frame, which are also machined. To adjust the pedestal horizontally, wooden keys of the necessary thickness are fitted between the surface *S* and the pedestal base. The height *H* is equal

to the highest point of the pedestal cap when raised clear of the cap-bolts CB + about 6″ to allow the engineer to remove the cap. The length l_1 is equal to l, the length of the base, + the amount of horizontal adjustment allowed on the pedestal

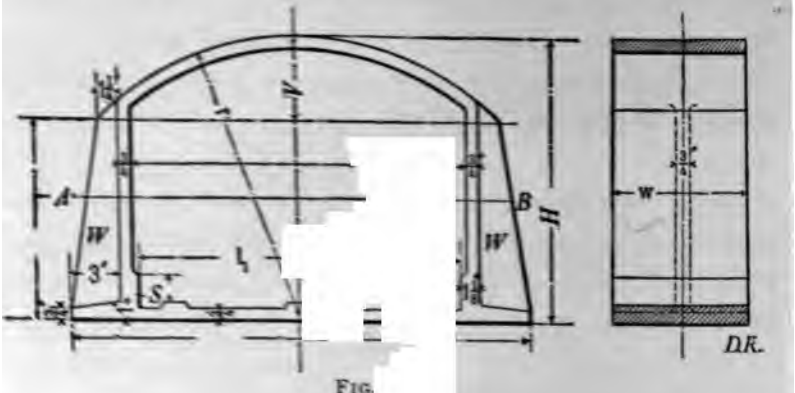


FIG.

+ ¼″. The width w is made . uit the thickness of the wall, which is usually built to average from 8″ to 12″. The proportioning of such a piece is largely a matter of experience, none of the parts being calculated for strength.

Exercise 69.—Draw a pedestal and wall box-frame of the designs shown in Figs. 148 and 150, placing the pedestal in position on the wall box-frame, to which it is secured by two square-headed bolts the heads of which project below the base. Make the pedestal to suit a shaft 2½″ in diameter, the length L equal to $3D$, and the width w of the frame equal to 8″. Show a **half-elevation** and **half-sectional elevation** of the pedestal, and an **elevation** of the wall box-frame, also a **plan** view of the pedestal with half of the cap removed, and in combination with this view show a section of the wall box-frame at the line AB. Make also an **end view** of the pedestal

and a **sectional end view** of the wall box-frame. All parts of the pedestal are proportional to the diameter D of the journal. Fill in all dimensions omitted. *Scale full size.*

Construction.—Draw the vertical and horizontal centre lines of the journal, then determine the distances from centre to centre of the bolts by drawing the line 1 which represents the top of the cap-flange, and the arc 2, which represents the top of the cap at the centre of the bearing. The centres of the cap-bolts can now be determined by making the corners of the nuts from $\frac{1}{16}''$ to $\frac{1}{8}''$ clear of the fillet which joins the lines 1 and 2. It is obvious that the bolts may be brought nearer together by either increasing the thickness of the cap-flange or cutting out the curve 2 around the nut, but on small pedestals for line shafting this is unnecessary. The radius r is made equal to half the distance across the angles of the nut $+ \frac{1}{4}''$ for finish. The distance from centre to centre of the holding-down bolts is equal to the distance $b +$ the horizontal adjustment (equal to the length of the hole $-$ diameter of bolt) $+$ the diameter of the washer $+$ the radii of the fillets, which may be made equal to about $\frac{1}{4}''$. Determine the radius r of the arched top of the wall box-frame by making V, the versed sine of the arc, equal to $\frac{e}{4}$.

Half the elevation is sectioned, to show more clearly the method employed to keep the Babbitt lining from turning with the shaft, the form of head on the cap-bolts, and also that the diameter of the holes through which the cap-bolts pass is greater than the bolt diameter. The plan view is shown with the cover removed from one side of the bearing, to show the form of that part of the bearing through which

the shaft passes. The fitting-strips on the under side of the base are of the same proportions as in the previous exercise. When practicable it is usual to provide the piece to which the bearing is fastened with fitting-strips also, as in Fig. 150.

Post Bearings.—When the bearing has to be secured to a vertical surface, the base is cast on the side, as shown in Fig. 151. In the design shown in Fig. 152 it is necessary to provide the cap with four bolts because of the webs *W*, which are in the way of the bolts being placed on the centre as in Fig. 148. The bearing is arranged in this case for two grease-cups, which are screwed to the cap at the tapped holes *O*. The cap-bolts are kept from turning when the nuts are being screwed down by projections *h* cast on the under side of the box.

Exercise 70.—Draw the elevation and an **end view half in section**, as shown in Fig. 152. Draw also a **plan view** of the top projected from the elevation. Make $D = 2\frac{3}{4}''$, and L = three times D. Parts not dimensioned are in the same proportion to D as in the preceding exercise. *Scale half size.*



FIG. 151.

Construction.—Draw the centre lines of the bearing, taking care to leave sufficient space to draw the plan. Mark off the distance that the bearing projects from the post, then determine the length and width of the base. **The centres of the bolts *PB* should be in a distance at least equal to the radius of the washer $+ \frac{1}{8}''$ from the ends of the base.**

FIG. 148.

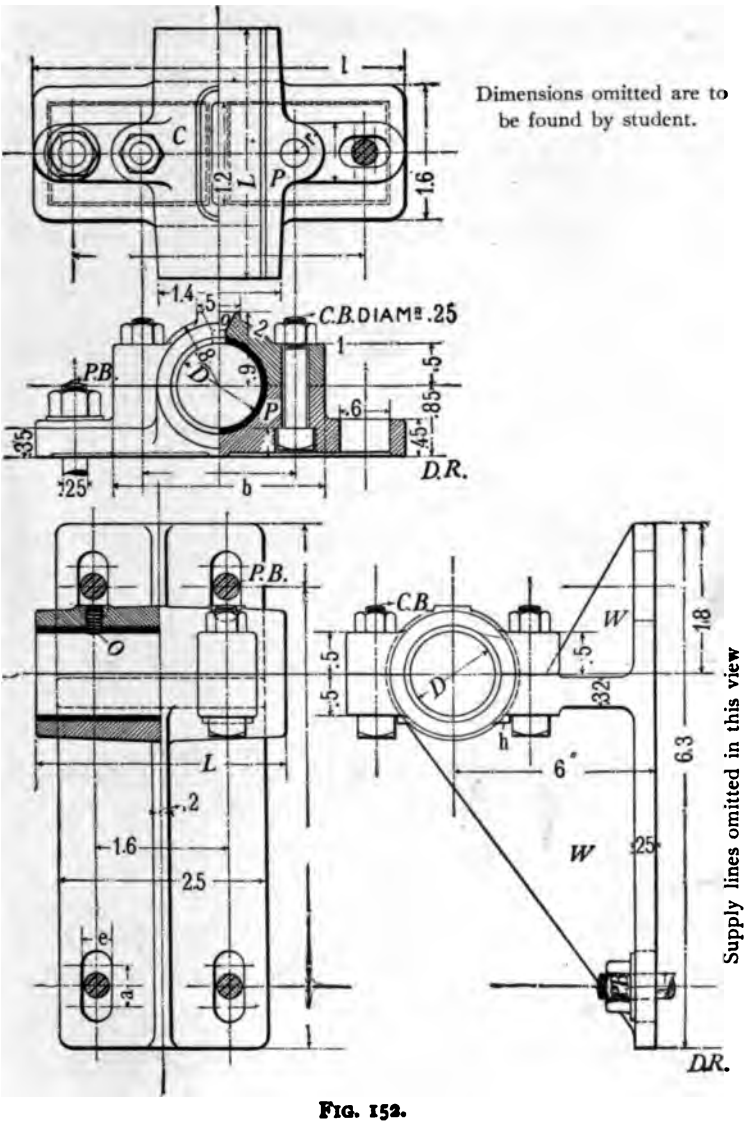


FIG. 152.

The vertical adjustment a is made equal to $1\frac{1}{2}''$. As the oblong holes are cored, the width e is $\frac{1}{8}''$ greater than the diameter of the bolts.

Wall Brackets are employed to carry pedestals which support a horizontal shaft running parallel and near to a wall. The bracket, Fig. 153, is fastened to the wall by means of three bolts which pass through it and the wall. The pedestal

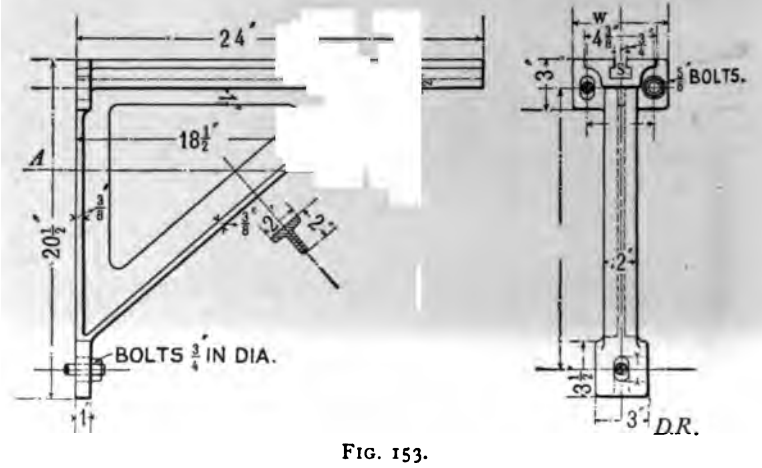


FIG. 153.

is secured to the upper surface by square- or T-headed bolts which slide in the I-shaped slot S which runs the whole length of the bracket. By this arrangement the distance that the pedestal is from the wall can be adjusted.

Exercise 71.—Draw a wall bracket to the proportions given in Fig. 153. Make the slot S suitable for a $\frac{5}{8}''$ square-headed bolt. Draw also a **section**, the plane of section passing through the bracket at the line AB. *Scale half size.*

SELF-ADJUSTING BEARINGS.

Bearings for supporting line shafting may be divided into two classes, Rigid and Self-adjusting. When shafting is supported upon a number of rigid bearings it is essential that they all be in line one with another, in order that the pressure be distributed over the entire surface of each. This is possible with bearings of the "rigid form" having comparatively long boxes when they are rigidly supported, but when supported upon insecure foundations, which are liable to sink, the bearing will assume such a position in relation to its journal as is shown in Fig. 154, where the entire load is carried

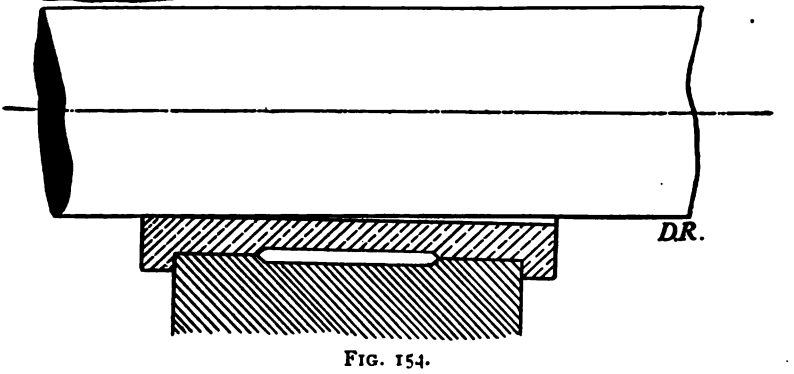


FIG. 154.

upon a small portion of the bearing. Such a condition exists also where the distance between the bearings is great in comparison with the shaft diameter, owing to the lateral deflection of the shaft by the gearing. Under such conditions the oil is forced out from between the rubbing surfaces, causing the metals to heat and seize by metallic contact.

To avoid this localization of pressure, bearings with a ball-and-socket joint are used, which to a limited extent adjust themselves to the various positions of the shaft, so that

the axis of the bearing will always coincide with that of the journal.

This form of bearing makes it practical to use a long box, thus keeping the pressure between the journal and bearing light enough to retain an unbroken film of lubricant between the rubbing surfaces. With these conditions the boxes may be made of cast-iron, which is the cheapest and, if well lubricated, the most desirable metal for the purpose. Many engineers, however, these boxes with a white metal which rapidly sts itself to any irregularities on the journal, fect bearing more rapidly than would be the cas[illegible] er material. Again, with the cast-iron box, should ricant fail and the metals come in contact, they and destroy the journal, while, under the same the babbitt metal would melt without materially injurin[illegible] shaft.

Drop Hanger-frame.—When a shaft is supported overhead and is not near a wall the bearings are carried upon a frame, called a hanger frame, which is secured to the ceiling girders. Two forms are used, the U form, which braces the bearing on both sides, as shown in Fig. 155, and the J, which braces one side only.

The objection to the U form as commonly made is the difficulty in getting the shafts in and out of the hanger. This has been overcome to some extent by making the hangers open at the bottom of the U, as it were, and connecting the sides with bolts.

The J form has the advantage of facilitating the mounting and dismounting of the shaft, but is liable to vibrate unless made comparatively heavy.

Fig. 156 shows a hanger made by the Dodge Manufacturing Co. which combines the advantages of both forms. This is attained by making the hanger open on one side and providing it with detachable links *L*, "which are split, and by bolts *LB*, drawn together upon taper cones *C*, cast on the

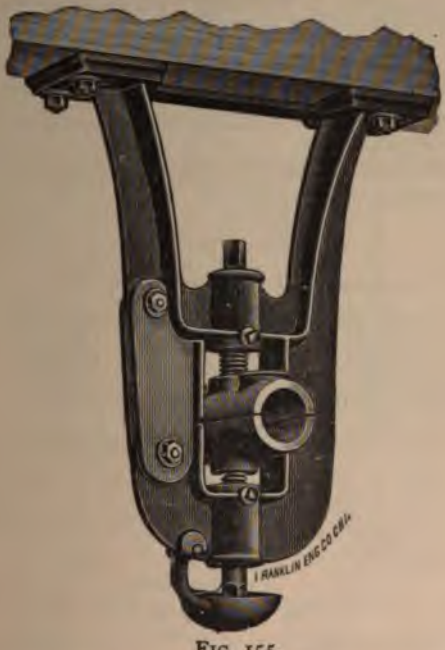


FIG. 155.

hanger frames *F*, which match corresponding recesses in the parts of the links. These links are thus drawn up to a positive bearing and form a connection which is virtually solid, and yet they are easily removed and replaced." Fig. 156 shows a shaft hanger with an adjustable bearing *B*, which is carried between the adjusting screws *P* and *P'*, called the plungers. These plungers are screwed into the frame *F* and serve a double purpose; first, they are a means of obtaining a vertical adjust-

ment; second, they provide the sockets, with which the spherical surfaces on the box engage, to form the ball-and-socket joint. The plungers are locked in position by the set-screws *S*. The bearings are lubricated by filling the cups

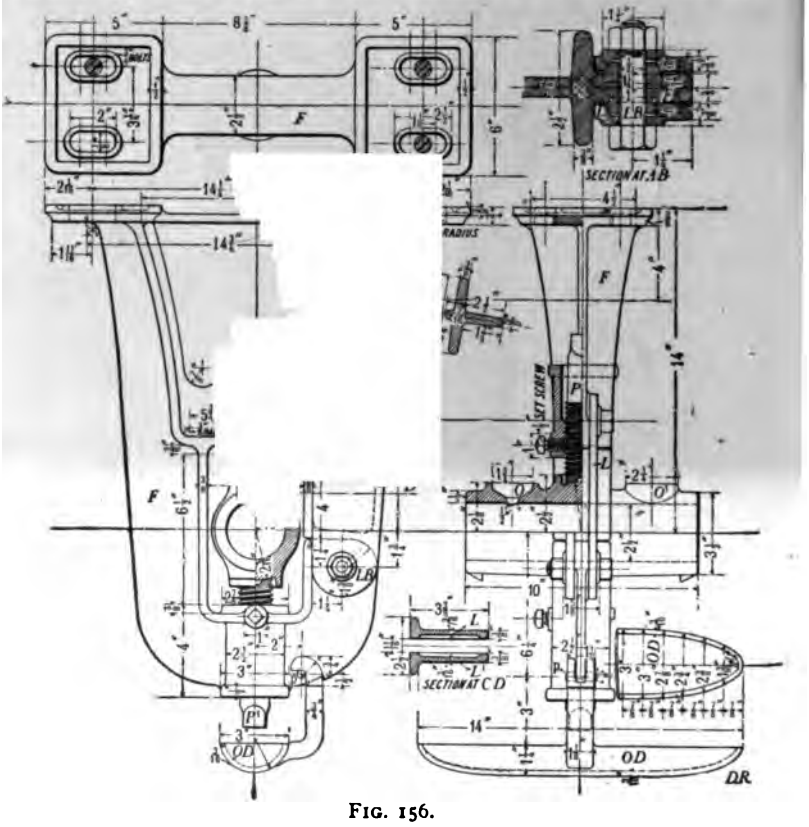


FIG. 156.

O and *O'* with grease, or cotton saturated with oil. The drippings of waste oil from the box are caught in the oil dish *OD* attached to the frame by hooking the head over the pin *P*, which is cast on the frame.

Exercise 72.—Draw the **front** and **end elevations** partly in

section, as shown in Fig. 156, a **half plan** and a **half-sectional plan** of the side to the right, the plane of section passing through the hanger at the centre line. *Scale half size.*

Draw also **full-size sections** of the frame, the plane of section passing through the hanger at the lines AB, CD, and EF.

Fig. 157 shows Sellers method of forming the ball-and-socket joint on adjustable hanger bearings. The plungers P and P' have shallow threads which extend along a portion of the plungers, while the threads in the boss are cut the entire length of the boss. The plungers are locked in position by the set-screws S, the points of which are made to press against the plain part of the plungers below the threads. The plungers are cast hollow, and are used as lubricators by filling them with cotton saturated with oil, which, under ordinary conditions, is sufficient to lubricate the journal. The openings O and O' are filled with tallow which is solid at ordinary temperatures but melts should the bearing become heated. The outer end of the plungers has a hexagonal hole to receive a key by means of which the screw is turned when adjusting the bearing.

Exercise 73.—Design a hanger-frame and bearing, altering the frame shown in Fig. 156 to suit the arrangement of plungers and bearing shown in Fig. 157, and design a method of fastening a drip-catcher to the frame, other than that shown in Figs. 155 and 156, which must be so arranged that it can be easily removed and replaced. Show a complete FRONT ELEVATION, SECTIONAL END VIEW, and PLAN projected from the front elevation.

Make $D = 2\frac{1}{2}''$, and length of bearing $= 4\ D$. *Unit of proportions is* $1.4\ D + .2$. *Scale half size.*

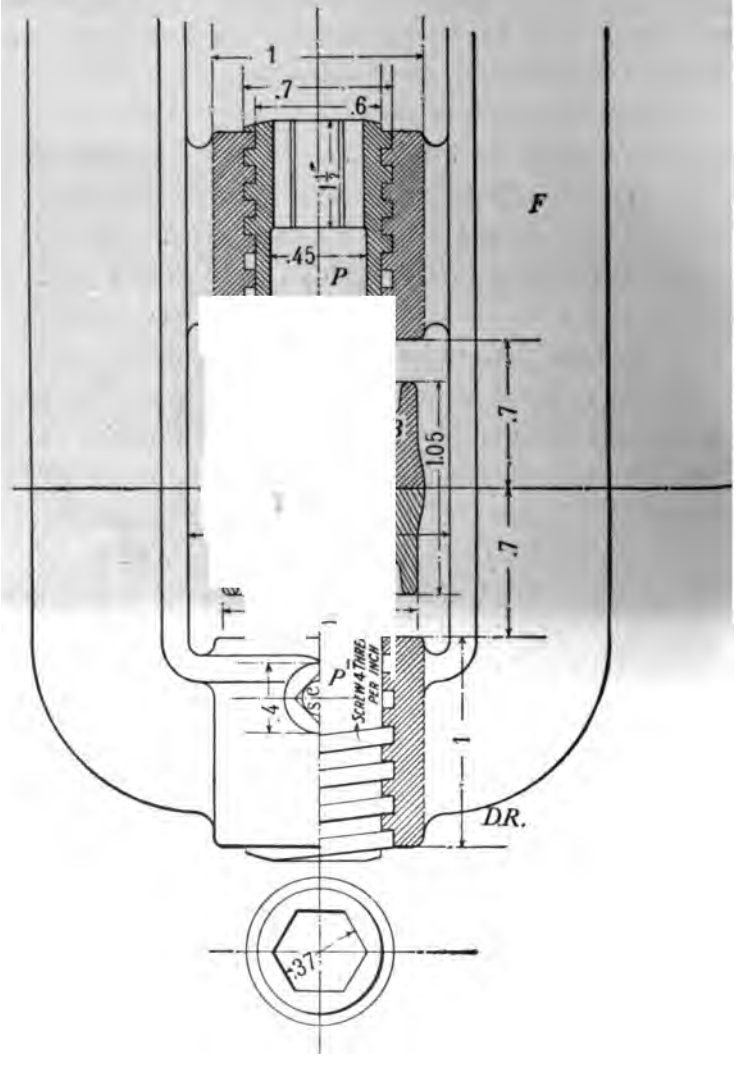


FIG. 157

Wall- or Post-hanger is employed to serve the same purpose as the Wall bracket with its separate pedestal. The frames of these hangers are designed on the same

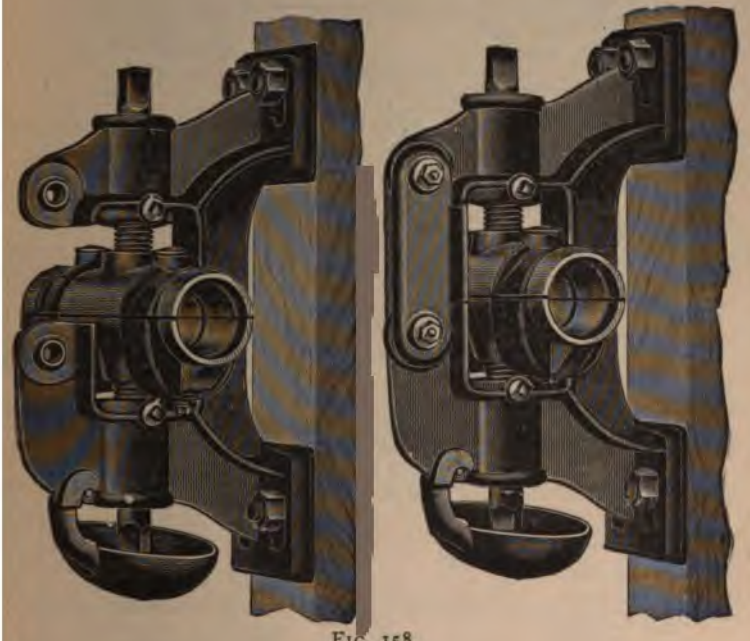


FIG. 158.

general lines and principles as the drop hanger-frames shown in Fig. 156. This hanger is shown in Fig. 158, with and without the double brace links, fitted with chain lubricating-bearings of the design shown in Fig. 160.

Exercise 74.—Draw FRONT ELEVATION and two END VIEWS as shown in Fig. 159, and a PLAN VIEW projected from the front elevation. *Scale 8″ to the foot.* Show also full-sized sections, the plane of section passing through the frame at the lines *AB*, *CD*, and *EF*.

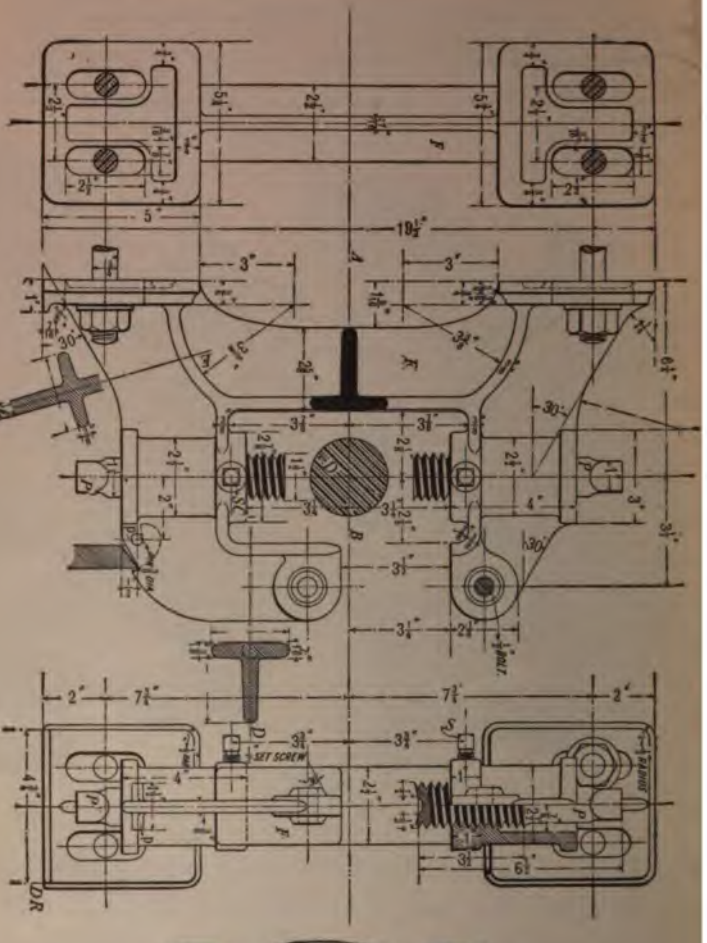


FIG. 159.



FIG. 160.

Chain Lubricating-bearing.—This type of bearing is designed to be lubricated by means of endless chains *C* which hang over the shaft, and as it revolves the chains revolve with it, passing through the oil in the reservoirs *O R* formed at each end of the box.

The chain *C* consists of a series of parallel links which form surfaces to which the oil adheres by capillary attraction, and is carried to the shaft, spreading through the channels *O C* to all parts of the bearing. All surplus oil falls back into the oil reservoirs, to be used again until it becomes thick or dirty, and is then drawn off by removing the plugs *S*.

Exercise 75.—Draw the chain lubricating-bearing shown in Fig. 161, showing a HALF ELEVATION and HALF SECTIONAL ELEVATION; an END VIEW projected from the right HALF SECTIONAL END VIEWS projected from the left-hand end, the plane of section passing through the bearing at the lines *AB* and *CD*, and a PLAN with half of the upper box removed. *Scale full size.* Draw also an ELEVATION AND PLAN of a part of the lubricating chain as shown in Fig. 163. *Scale four times full size.*

Construction.—Fig. 162 shows a method of finding the centres of the chain represented in the end-view in position on the shaft. In this construction the centres may be taken on the curve unless from the points 1 to 2, where the radius is small. At this part step off chords equal in length to the pitch of the chain, and, parallel to the chords, draw lines tangent to the arcs. The intersection of the tangent lines may be taken as the centres of the chain at that part.

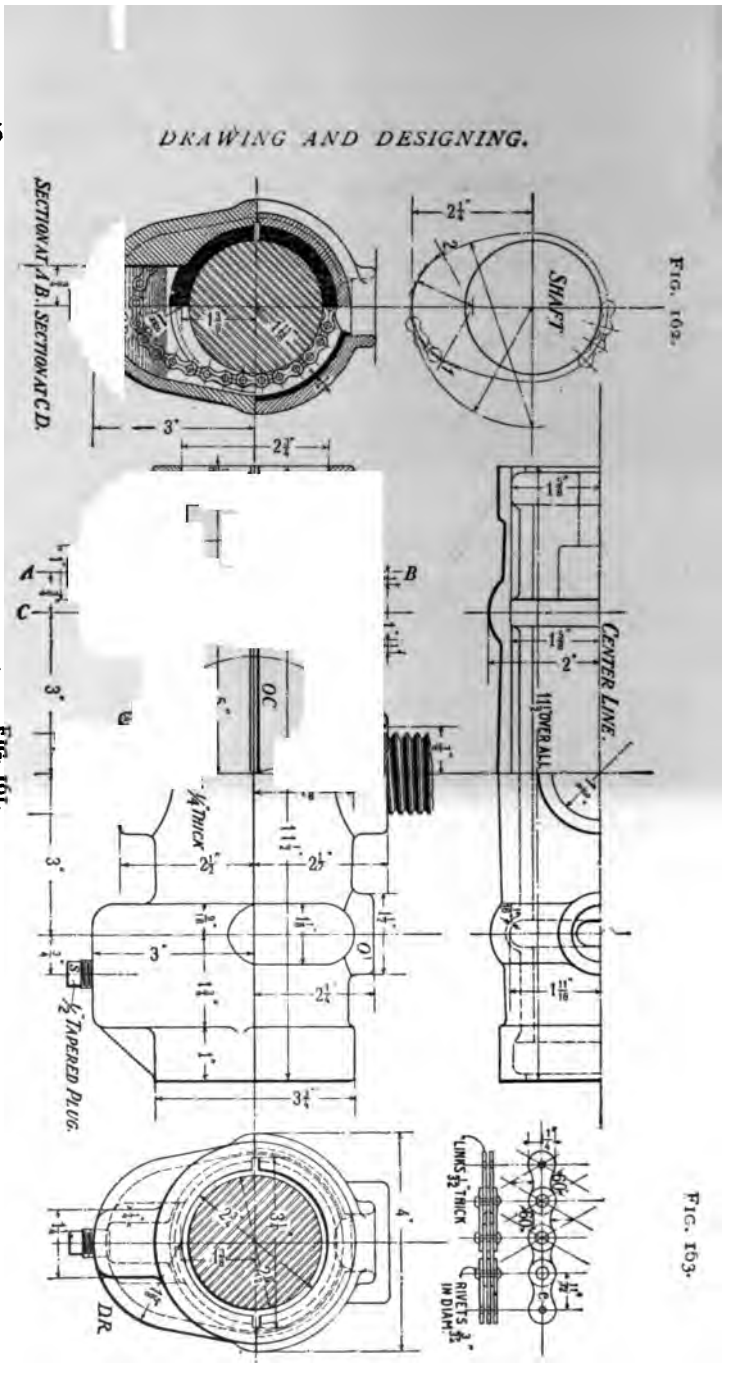


FIG. 161.

FIG. 162.

FIG. 163.

Bushes, Steps, or Brasses are names given indiscriminately to the bearings proper, i.e., the brass or bronze parts, that are in contact with and support the journal. They afford a means of taking up the lost motion due to wear, thus insuring that the journal with which they engage will have the required motion about the given axis. They must be made of a material that will allow the journal to run in contact with it with a minimum amount of friction, and will withstand wear without wearing the journal. They must also have sufficient strength to resist the stresses that come upon them, without undue yielding. When supporting a wrought-iron or steel shaft, gun-metal, to a limited extent, fulfils all these requirements. Other metals possess some of these qualities in a higher degree without having them all.

White metals, such as "babbitt's" or "magnolia" metals, offer less frictional resistance, and their surfaces may be destroyed without injuring the surface of the journal (as would be the case with the bronzes), but they are too soft to be used alone unless subjected to an exceptionally light load. The position of the bush in the supporting frame depends upon the direction of the pressure. In the majority of bearings the resultant pressures are in one or two directions, and all lost motion can be taken up by making the bearings in two parts. The ordinary forms of two-part bearings are shown n Figs. 164 to 167. The forms shown in Figs. 164 and 165 are turned, and the supporting frame is bored with a cylindrical hole into which the bearings are fitted. To prevent these forms from rotating with the shaft they are provided with rectangular lugs *L*, as in Fig. 165, or with steady pins *P*, as in Fig. 164.

FIG. 164.

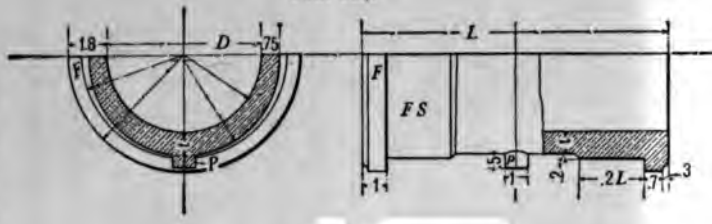
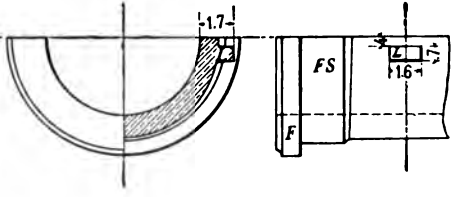
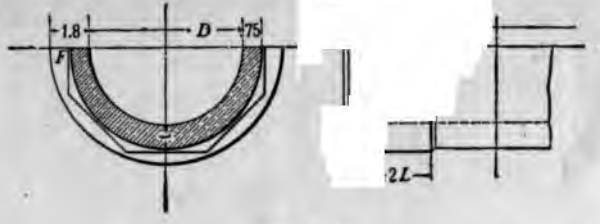


FIG. 165.





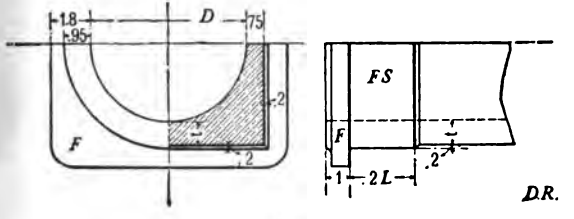


FIG. 166.

FIG. 167.

Unit = $t = .08D + .15$

D.R.

The pins may be either cast with the bush or driven in. The forms shown in Figs. 166 and 167 are cast square or octagonal and planed to fit correspondingly shaped surfaces in the supporting frames. The square form is the cheaper, but should it become hot it is liable to be distorted, owing to the unequal distribution of metal. To facilitate fitting, and reduce machining on bearings, it is usual to support them at their ends only, by forming projecting faces *F S* at each end. This may be done successfully on small bearings subjected to a steady load, but on crank-shaft bearings it is advisable to support them over their length. The bearings should be divided on a line normal to the resultant pressures and, as they will wear very little at that part, they may be made thinner than at the part where the pressure is greatest. To keep the bearings from moving laterally along the shaft they are provided with flanges *F*, between which the supporting frame fits, as shown in Fig. 169.

Sole-plates.—When a pedestal is secured to masonry or brickwork it is necessary to spread the pressure upon the journal over a large surface. For this purpose a Sole- or Base-plate is employed. These usually consist of a flat cast-iron plate with a bevelled surface upon which the pedestal can be adjusted horizontally by means of the wood keys *K*, which are driven in between the joggles *J* and the ends of the pedestal base, as shown in Fig. 169. The pedestal is fastened to the sole-plate by the bolts *P B*, which pass through it and the base of the pedestal. The sole-plate is secured to the foundation by the bolts *F B*. The width (*b*) of the sole-plate should be equal to (*a*) width of pedestal base + the amount of movement of pedestal along shaft + say $\frac{1}{8}''$.

Adjustable Base-plates are used for adjusting bearings vertically and horizontally. The vertical adjustment is made by sliding wedges which may be arranged either laterally (as in Fig. 168) or longitudinally. The horizontal adjustment is



FIG

effected by means of set-screws which take the place of the wooden keys shown in Fig. 1

Pedestal or Pillow-block Bearings are used where it is necessary to have a bearing that is rigid and yet adjustable. Fig. 169 shows the ordinary form of pedestal bearing employed for supporting shafting from 3″ to 8″ in diameter. The inner surfaces of the block *P* and cap *C* are formed to suit the outer surface of the bushes. When the block is prepared by hand-work to receive the bushes it is provided with fitting strips *F S* to facilitate fitting, but when prepared by planing, the strips are unnecessary. Some engineers make the bushes that they do not touch each other when the shaft is in position, and as the bushes wear, a space being left between the cap and the pedestal, they are brought nearer together by screwing down the cap *C* by means of the bolts *C B*. To keep the cap from being screwed down too far, causing the bushes to bind the journal, the space between the cap and the pedestal is sometimes filled with hard wood and the wear is taken up by filing down the hard-wood distance-

pieces, thus allowing the cap to be screwed down a limited distance. Others make the bushes in contact with each other, as in Fig. 169, when the bushes fit the shaft, and when they become worn they are filed down sufficiently to compensate for the wear. When the bushes do not come in contact with each other and no distance-piece is used, the cap-bolts should be provided with double nuts. After the pedestal has been

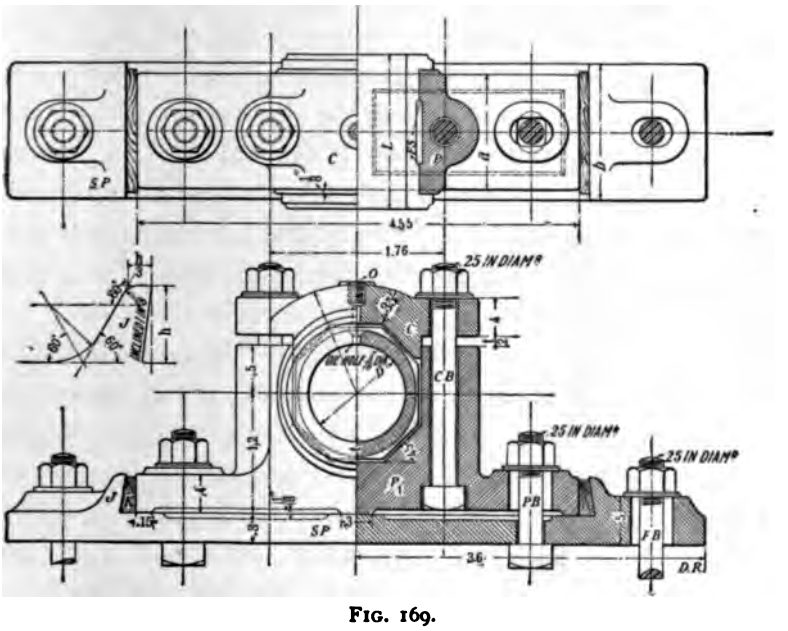


FIG. 169.

adjusted to suit the shaft, it is held in position by the bolts *P B*. The holes in the base- and sole-plate through which the bolts *P B* pass are made oblong to allow the pedestal to be moved along the shaft or transversely to it.

To facilitate the fitting of the pedestal to the piece upon which it is carried, the base is provided with fitting-strips around the edges and across the centre. The oil-cup is usually

cast with the cap *C*, or screwed into the tapped hole *O*, Fig. 169. On pedestals having journals less than 3″ in diameter *O* may be made to receive an oil-cup with a ¼″ pipe tap-shank, and when over 3″, with a ⅜″ pipe tap-shank.

Exercise 76.—Draw a general arrangement of a pedestal and sole-plate, Fig. 169, substituting the form of bearing shown in Fig. 164. Show a HALF ELEVATION and HALF SECTIONAL ELEVATION, the plane of section passing through the centre of the [illegible] HALF PLAN and HALF SECTIONAL PLAN, the [illegible] passing transversely through the centre of the [illegible] he elevation project a HALF END-ELEVATION [illegible] H. [illegible] ONAL END-ELEVATION, the plane of section passi[illegible] the centre of the pedestal. Make the length of the h[illegible] gh the sole-plate and pedestal-base sufficient to all[illegible] destal to move ¼″ in either direction. Make $D = 4'$ and [illegible] $= 2D$. *Scale half size.*

Construction.—All parts dimensioned in decimals are in terms of *D* (the diameter of the journal). Parts marked in inches are constant. Any parts not dimensioned can be determined by the student from knowledge derived from previous exercises. A method of drawing the joggles *J* is shown at Fig. 169, which will be readily understood from the drawing.

SELF-LUBRICATING PEDESTAL.

In this design, Fig. 170, an oil reservoir *OR* is formed on the under side of the bearing, in which loose rings *R* are revolved by their friction on the journal, thereby raising a continuous supply of oil to the upper side of the bearing, thus keeping the journal thoroughly lubricated and not

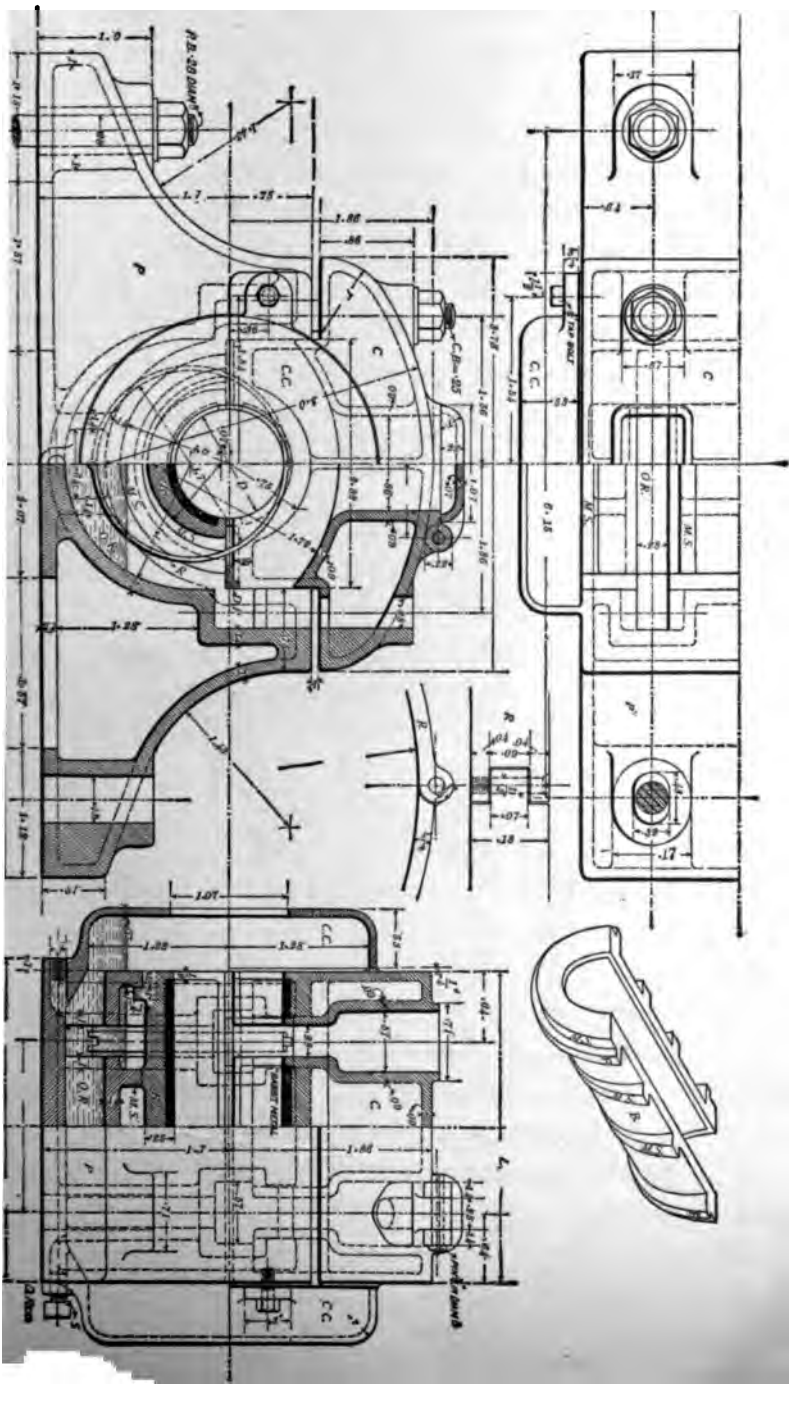


FIG. 171.

wasteful, as the surplus oil that flows out of the bearing is caught in the chambers *CC* and carried back to the reservoir *OR*.

As the same oil, in this form of lubricator, is being used repeatedly, after a time it becomes dirty and thick and is then useless. By removing the screws *S* the old oil is drained off, and the reservoir can then be replenished by pouring new oil into the openings in the cover. These openings are made large, so that the e[illegible] if the rings are revolving.

This pedestal is a[illegible] *down* pressure, and as there will be very little wea[illegible]per bush it is cast with the cap *C*. The lower [illegible]eparate piece, as shown by the sketch, Fig. 171. [illegible]e the machining it is provided with projecting f[illegible], called machining strips, which fit upon correspond[illegible]rojections on the pedestal, and are made concentric with [illegible] shaft, so that to remove the bush it is not necessary to withdraw the shaft, as the bush when relieved from the load can be turned to the upper side of the journal.

By this arrangement the pedestal is practically independent of wear, as the bushes can be removed and re-babbitted with little trouble or expense.

To keep the bush from moving laterally, flanges *F* are cast at each end which fit inside of the end machining strips on the pedestal.

The lower bush is kept from turning by the distance piece *DP*, which also keeps the cap from being screwed down too far and clamping the shaft. To take up the wear of the bushes, the distance pieces *DP* are planed to let the cap go further into the pedestal. To allow this, a space

A is left between the pedestal and the cover. This space need not be greater than the thickness of the babbitt lining, which should be from $\frac{3}{8}''$ to $\frac{1}{2}''$ thick.

The cap is made to fit into the pedestal so as to sit squarely upon the journal, and does not depend upon the cap bolts to prevent lateral movement.

The cap is usually held down by two bolts, but to avoid large bolts in the larger sizes of pedestals it is quite common practice to use four. The bolts in this case are made square in section, and have T heads which fit into recesses cast in the pedestal. The pedestal is held in the proper position by the bolts *PB*, which pass through oblong holes in the pedestal to allow for longitudinal adjustment in either direction. This form of pedestal is suitable for journals from 5″ in diameter up.

Length of Bearings.—The frictional resistance at the surface of the journal converts the mechanical energy into heat, and, unless the area of the journal is sufficiently large to allow the heat to radiate as fast as it is generated, the temperature will become great enough to destroy the lubricant, allowing the rubbing surfaces to come in contact and adhere to each other. The radiating surface would be enlarged by increasing the diameter of the journal, but the velocity of the rubbing surfaces would also be increased; therefore the frictional resistance and the space through which it acts would be greater. Thus it will be seen that to add to the radiating surface without increasing the work at the surface of the journal we must increase the length of the bearing.

In a paper read before the Manchester (England) Associa-

tion of Engineers, Professor Goodman stated that the area of a bearing should be such that not more than one thermal unit of heat is generated per square inch of bearing surface per minute.

Let P = total pressure in pounds;

μ = coefficient of friction;

S = speed of circumference of journal in feet per minute $= \frac{\pi DN}{12}$;

N = number of revolutions per minute;

A = area of bearing in square inches, i.e., the diameter $W \times$ the length L;

D = diameter of journal in inches;

L = length of journal in inches;

W = width of the chord in contact, in inches.

Foot-pounds of work done per minute at the circumference of the journal $= P\mu S$. The thermal units per minute $= \frac{P\mu S}{778}$, and $A = \frac{P\mu SW}{778D}$, from which $L = \frac{P\mu S}{778D}$, in inches.

With steel journals running in bronze or white-metal bearings, having continuous lubrication, μ, the coefficient of friction may be taken at .0056.

Exercise 77.—Design a self-lubricating pedestal for a shaft 6″ in diameter, of the form shown in Fig. 170, to carry a load of 35,000 pounds, and run at a speed of 300 revolutions per minute.

Show a HALF ELEVATION, HALF-SECTIONAL ELEVATION, the plane of section passing through the centre of one of the lubricators, a HALF END ELEVATION and HALF TRANSVERSE SECTION, the plane of section passing through the pedestal at the centre, a HALF PLAN of the left-hand side of the pedestal, a QUARTER PLAN with the cover (C) removed, a QUAR-

ER-SECTIONAL PLAN, the plane of section passing through ıe centre of the shaft. *Scale* 3″ *to the foot.*

Make also full-size drawings of the lower bush, showing a ALF ELEVATION and HALF-SECTIONAL ELEVATION, a HALF ND VIEW, and a HALF TRANSVERSE SECTION, and a plan nd elevation of the ring-joint as shown.

All points are proportional to the diameter (*D*) of the ournal, except those parts which are constant for journals f various sizes.

CHAPTER VII.

BELT GEARING.

Belts.—Amc[illegible]erent kinds of material used for belting are [illegible] gutta-percha, India-rubber, canvas, camel-hair, [illegible] wire or hemp rope, steel bands, flat chains,

The most common [illegible]al practice are leather and cotton, the latter oft[illegible]ated with India-rubber and known as gum belts.

Leather is more dur[illegible] gum under most conditions, but for main driving the latter is superior, having an adhesion which is claimed to be one third greater than the former.

Transmission of Motion by Belts.—Motion may be transmitted from one pulley to another with uniform linear velocity by means of a belt, provided there is no slipping of the belt on the pulley; i.e., regarding the belt as inextensible every part of it will have the same velocity as the outside rim of the pulley.

Referring to Fig. 171, let d_1 and d_2 be the diameter of the driver and driven pulleys respectively, and let N_1 and N_2 be their revolutions per minute and V the velocity of the belt.

The speed of the rim of the driver

$$= d_1 \pi N_1 = V \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

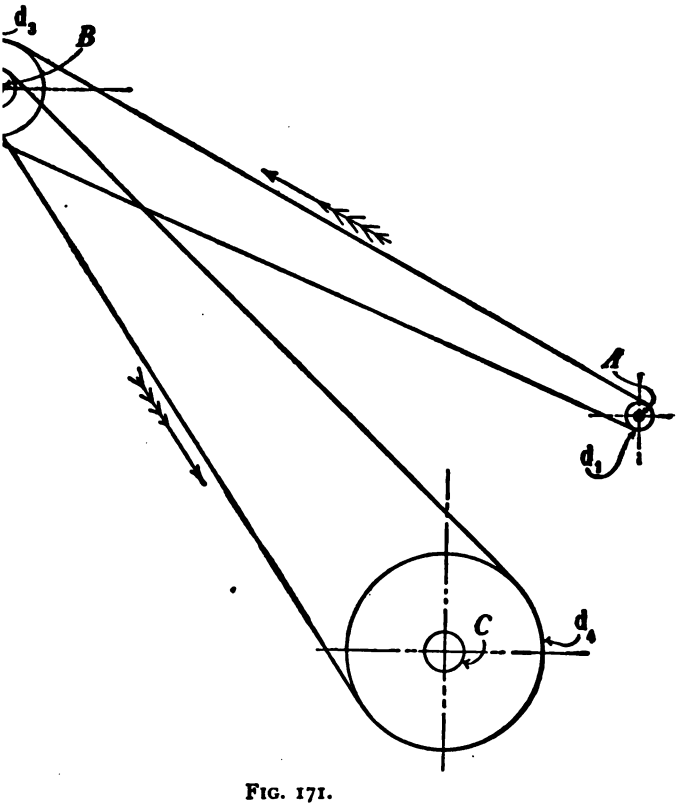


FIG. 171.

speed of the rim of the driven

$$= d_2 \pi N_2 = V \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

e

$$N_1 = d_2 \pi N_2 \quad \text{or} \quad d_1 N_1 = d_2 N_2 \quad \text{or} \quad \frac{N_1}{N_2} = \frac{d_2}{d_1}. \quad (3)$$

ll questions concerning the velocity ratio of belting
:y diameters should be taken to the centre of the belt

thickn hus the virtual diameter of the pulley would be the nor l diameter plus *one thickness* of the belt. For other calcu lations the thickness of the belt is so small it may be ne; ected without much error.

Exan 1.—In the draughting-room at Sibley College there is a valve-motion model driven by an electric motor. The shaft *A* of the motor carries a pulley 1½″ diameter from which passes a belt to a 15″ pulley on a counter-shaft *B*. This shaft car 5″ in diameter connected by a belt to the ey of 30′ diameter on the valve-motion m

The speed of o R. P. M. Find the speed of the valve-motio P. M., Fig. 171.

From formu

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} \quad \therefore \frac{1450}{N_2} = \frac{30}{6} = 50.$$

Substituting we get

$$\frac{1450}{N_2} = 50 \quad N_2 = \frac{1450}{50} = 29 \text{ R. P. M.}$$

Some Practical Rules.—The width of belts should be about 25 per cent less than the face of the pulley.

It has been demonstrated by experience that large pulleys and fast-running belts are much more economical than small pulleys and slow-speed belts. All pulleys should be carefully centred and balanced on the shaft. Driving-pulleys carrying shifting-belts should have a perfectly flat surface. All other pulleys should have a convexity of $\frac{1}{8}$″ to 12″ of width; when curved the chord of the arc should be the same. For

pulleys smaller than 12″ wide, from $\frac{3}{8}$″ to $\frac{1}{2}$″ per foot of width should be used.

Pulley diameters should be as large as can be used provided the belt speed is kept within 5000 feet per minute, which is held to be the limit of speed for belt economy.

With regard to the position of idle pulleys in relation to the driving-pulley Taylor says, "Idle pulleys work most satisfactorily when located on the slack side of the belt about one quarter away from the driving-pulley."

Transmission of Power by Belts.—Let two pulleys A and B be connected by a belt with a tension equal to T_2. Until force is applied at A tending to produce rotation of the pulleys, the tension T_1 and T_2 will be equal; but as the force at A increases the tension in T_1 will increase, and that in T_2 will decrease until $T_1 - T_2 = P =$ resistance to rotation at the rim of the pulley; i.e., when the belt is at the point of slipping, the ratio of T_1 to T_2 will be a maximum and $= efa$, or $T_1 \div T_2 = efa$. Where e is the base of the Naperian system of logarithms, f is the coefficient of friction $= .3$, a is in π measure and $= a$ in degrees $\times$ 0.0174.

By logarithms we find that $T_1 \div T_2 = efa =$ log. $T_1 \div T_2 = fa$ log. $e = .4343fa$.

Example 2.—A six H.P. dynamo is to have a speed of 1450 R. P. M and has a 6″ pulley on its shaft. Power is obtained from an engine fly-wheel running at 58 revolutions per minute. To obtain the required velocity ratio between the engine and dynamo, the diameter of the fly-wheel will have to be 25 times that of the dynamo pulley with direct connection; but such a diameter would be practically impossible, so it will be necessary to install a counter-shaft. Let 18″ be the

mos le diameter for the largest pulley on the counter-shaft, the necessary speed of the counter-shaft will be $= 14 \quad \frac{6}{18} = 483$ R. P. M. Between the engine and counter-sh t the pulley diameter ratio $= \frac{483}{58} = 8.32$. Let the diameter of the fly-wheel be 50″ then its connecting-pulley on the counter-shaft will be $\frac{50}{8.32} = 6''$ nearly.

To determin t necessary to connect the dynamo with th : will have to find the value of $T_1 =$ to the .. the lower side of the belt.

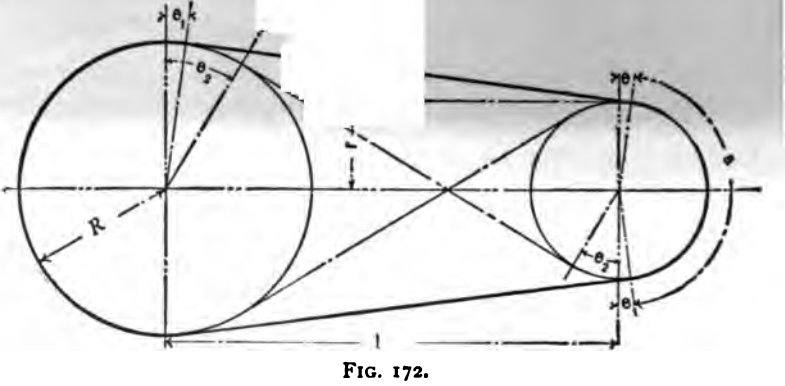


FIG. 172.

First find the work done by the dynamo $= 6 \times 33{,}000 =$ 198,000 foot-lbs. per minute; the rim of the dynamo pulley runs at $\frac{6\pi}{12} \times 1450 = 2277$ feet per minute; therefore $T_1 - T_2 = \frac{198000}{2277} = 87$ lbs. Let the centres of the dynamo shaft and counter-shaft be 15 feet apart, then (see Fig. 172)

$\tan. \theta = \frac{R - r}{l} = \frac{9'' - 3''}{180} = .04$, and from a table of natural trig. functions we find that tan. .04 = 2.25°.

$a = 180° - 2\theta = 175.75$, a in π measure = 175.75 × 0.0174 = 3.05. Then log. $T_1 \div T_2 = .4343 \times .3 \times 3.05 = .3974$; from a table of logarithms we find that .3974 is the log. of the number 2.50, therefore

$$T_1 \div T_2 = 2.50.$$

Combining these equations thus:

$2.50\ T_1 - 2.50\ T_2 = 87 \times 2.5''$, $T_1 - 2.50\ T_2 = 0$,

$1.50\ T_1 = 217.5$, we find $217.5 \div 1.50 = 145$, and allowing 70 lbs. per inch width of belt, then

$$145 \div 70 = 2.06, \text{ say } 2\tfrac{1}{8}''.$$

Some Practical Rules for the Transmission of Power.—Richards gives the following rule for the size of driving-belts, which he says is near enough for all cases that arise in ordinary practice.

$$\text{H.P.} = \frac{V \times W}{A}. \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Where V = the velocity of the belt in feet per minute.

W = the width of the belt in feet.

A = the area given to suit different conditions in the following table:

TABLE 30.

LEATHER BELTS SINGLE THICKNESS. 1 H. P.	GUM BELTS AVERAGE THICKNESS. 1 H.P.
On smooth iron pulleys.....80 ft.	On smooth iron pulleys.....60 ft.
On wooden pulleys.....65 ft.	On wooden pulleys.....50 ft.
On covered pulleys.....50 ft.	On covered pulleys.....35 ft.

B uld be made as wide as possible; they are often too n but never too wide.

T ss of Belts.—As belts increase in width their thickn ould also increase. Double belts should be used on pu ver 12″ diameter. Large belts running at very high sp as in electrical work, should have slots punched through them in such manner and position as to prevent air cushion.

The follo thickness of belt and corresponding work l on a safe working stress of 320 lbs. per sq. ts, are given by Unwin:

Thickness of belt.....	$\frac{5}{16}$″	$\frac{3}{8}$″	$\frac{7}{16}$″	$\frac{1}{2}$″	$\frac{9}{16}$″	$\frac{5}{8}$″	$\frac{11}{16}$″	$\frac{3}{4}$″
Working tension in lbs. per inch of width....	60	120	140	160	180	200	220	240

For other rules and formulæ see **Kent's Engineers' Pocket** Book, page 876.

For a safe working tension under ordinary conditions, many authorities allow only 45 lbs. per inch of width; but according to Mr. A. W. Smith, experiments have shown that a safe tension of 70 lbs. may be had per inch of width of belt.

Proportions of Pulleys (Figs. 173 and 174).—

a = centre of set-screw from end of hub = $1\frac{1}{4}d_1$.

a_1 = centre of bolt from edge of flange = $1\frac{1}{4}d_2$.

b = width of belt—see Example 2.

B = pulley face = $\frac{9}{8}(b + 0.4)$. (Unwin) (5)

d = shaft diameter.

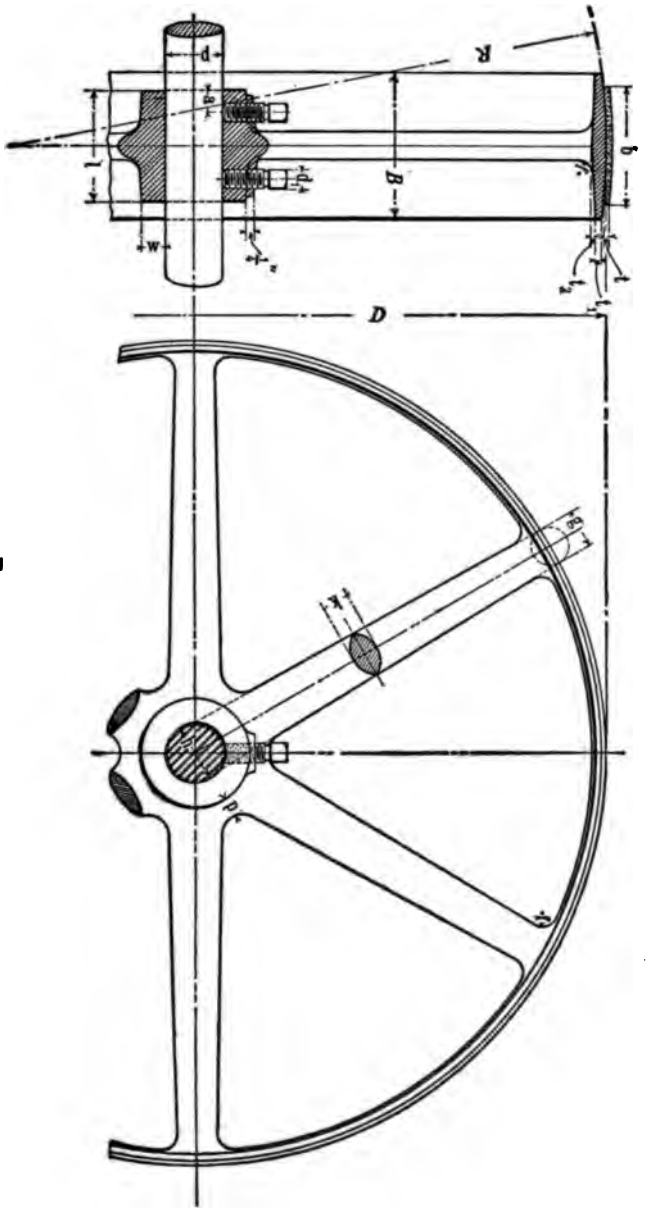


FIG. 173.

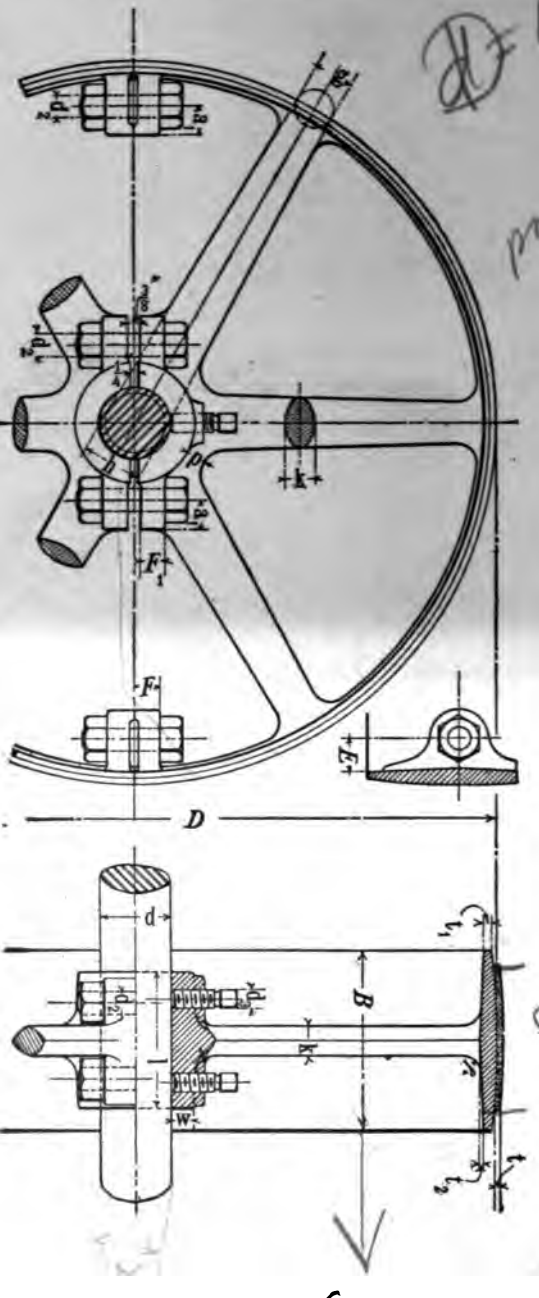


FIG. 174.

= diameter of set-screw in solid pulley $= \frac{1}{8}d + \frac{7}{16}''$. . (6)

= diam. of bolt in split pulley at rim and hub = eq. (6)

= set-screw for key $= .25d$.

= diameter of pulley.

= centre of rim bolt from inside of rim $= d_2 + t_2 + \frac{1}{4}''$. (7)

= radius at end of arms $= \frac{K}{2}$

$= d_2 + \frac{1}{8}''$. $F_1 = F$.

= width of arm at rim $= \frac{3}{8}h$.

= width of arm at centre of pulley

$$= \begin{cases} .6337\sqrt[3]{\frac{BD}{n}} \text{ single belt.} \\ .798\sqrt[3]{\frac{BD}{n}}. \end{cases} \quad \text{(Unwin.)} \quad \ldots\ldots\ldots (8)$$

= thickness of arm $= \frac{h}{2}$.

= length of hub $= \frac{1}{8}B$ to B.

= number of arms $= \frac{D}{36} + 4$. The nearest number divisible by 2 should be taken.

= thickness of rib surrounding hub between arms $= .31\ d$.

= thickness of belt—see Table 31.

= thickness of rim $= .6t + .005\ D$. (9)

= inside taper of pulley rim $= t_1 \div 2$.

$$= \text{thickness of hub} = \begin{cases} .14\sqrt[3]{BD} + \frac{1}{4} \text{ for single belt.} & (10) \\ .18\sqrt[3]{BD} + \frac{1}{4} \text{ " double " } & (11) \end{cases}$$

= radius of pulley crown = from 3 to 5 b.

Exercise 84.—A fan revolving with a speed of 1800 rev. min. develops 8 H.P. and has an 8″ pulley on its shaft. wer is obtained from an engine fly-wheel running at 75 . per min. Diam. of fly-wheel = 5 feet. Determine the per diameters of the intermediate pulleys and make a suit-

able working drawing of the largest of them, similar to Fig. 173 or Fig. 174. *Scale 6″ = 1 foot.*

See Example 2, p. 241.

Wood-split Pulley (Fig. 175).—The Committee on Science and the Arts of the Franklin Institute, in reporting on the Dodge Wood-split Pulley with wooden bushings, stated that in most cases wood-split pulleys are better than iron pulleys. Some of the reasons given for this are as follows:

(1) They are lighter than iron pulleys, lessening the weight on the line shaft and bearings and reducing friction.

(2) The compression fastening of the wooden pulley on iron or steel shafts with wooden bushings will hold the pulley on the shaft quite firmly, dispensing with the use of keys.

(3) The grip of a belt on a wooden pulley exceeds that on an iron pulley to an amount equal to at least 33 per cent.

(4) The method of fastening the wooden pulley to the shaft neither mars nor weakens the shaft, and prevents any tendency to throw the pulley out of balance, as is the case when keys and set-screws are used.

Construction.—"They are built of wooden segments, the face being made of poplar. The two halves of the pulley are secured to the shaft with bolts. The bushings to fit different-sized shafts are made of hard wood, thoroughly air-dried, then bored and kiln-dried; then each bush is counterbored to exact size of shaft, then carefully turned on the outside to fit the bore of the pulley. They are then cut transversely in halves."

Exercise 85.—Make complete working drawings of a wooden split loose pulley 14″ diam., shaft 2″ diam. Projections to be the same as shown in Fig. 175. *Scale 9″ = 1 foot.*

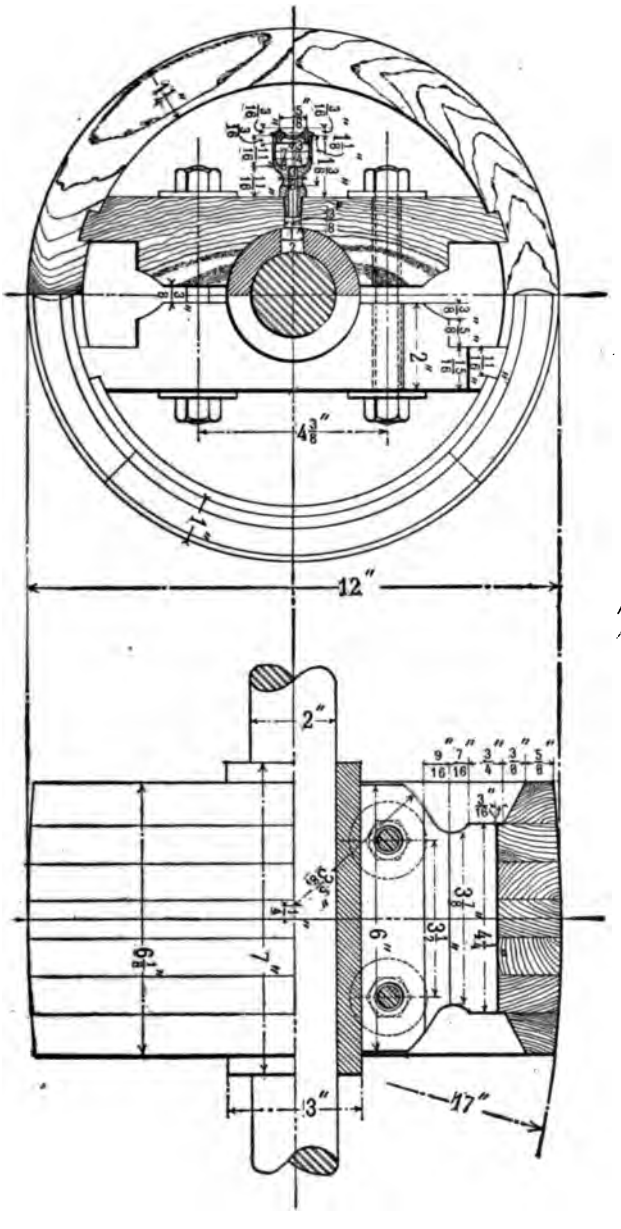


FIG. 175.

All-wrought-steel Pulley.—This pulley as manufactured by the Am. Pulley Co. is shown in Fig. 176. In a paper on the subject by Mr. E. G. Budd before the Franklin Institute in June 1897, the following advantages are claimed for the all wrought-steel pulley:

(1) They can be used in the heaviest service, clamped to the shaft without keys or set-screws, and never show a sign of slipping.

(2) There is no machining required. The rims and arms



FIG. 176.

are cut with shears and pressed into shape with hydraulic pressure.

(3) Economy of material and symmetry of form, requiring
› counterbalance.

(4) Being made of the best and strongest material,
is fully as light as the wood pulley, and much more
ırable.

Construction.—Referring to Fig. 176 it may be seen that
ıe rim is made up of four segments. It is divided once
ansversely and once longitudinally. The flanges on the
m at the centre of the face give a means of fastening it to
ıe arms. The rim edges are rolled, giving a neat appearance
ıd preventing the scraping of the belt in throwing it off or
ı.

The hub is made of half cylinders of heavy steel, and is
›nnected to the rim by a spider divided into four parts, two
ırts to each half of the pulley. The spider arms are flat
ıd have the edges lying in the direction of rotation. The
anner of fastening the arms to the hub and rim, and their
›rrugated section, as shown at *A* in Fig. 177, make them ex-
ptionally strong for their purpose.

Exercise 86.—Make a true working drawing of the all-
rought-steel pulley shown in Fig. 177 to the dimensions
ven. *Scale* $4'' = 1$ *foot.*

Cone-pulleys.—In operating machine tools it is often
:cessary to change power and speed. This is accomplished
ost easily by means of cone-pulleys. The driven pulley
ıs a series of steps whose diameters are proportioned so that
ıe belt shall fit all pairs of steps with an equal tension, and
hen the belt is shifted from one pair of steps to another
ıe velocity ratio will be changed.

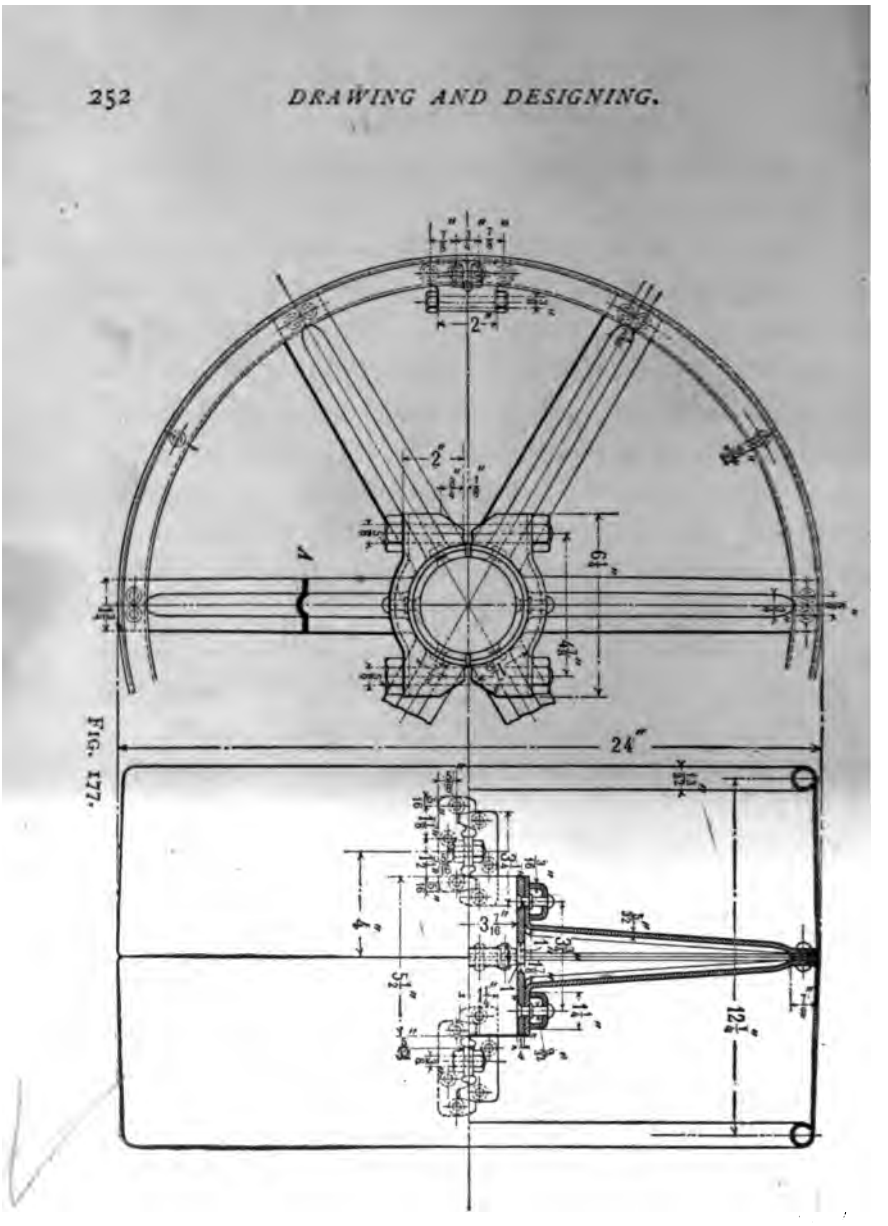


FIG. 177.

Length of Belts (Fig. 178).—

Let L = length of belt;

D = diam. of large pulley;

d = diam. of small pulley;

l = distance between centres of pulleys;

θ = angle whose sine = $\frac{D+d}{2l}$ for crossed belts and

$\frac{D-d}{2l}$ for open belts.

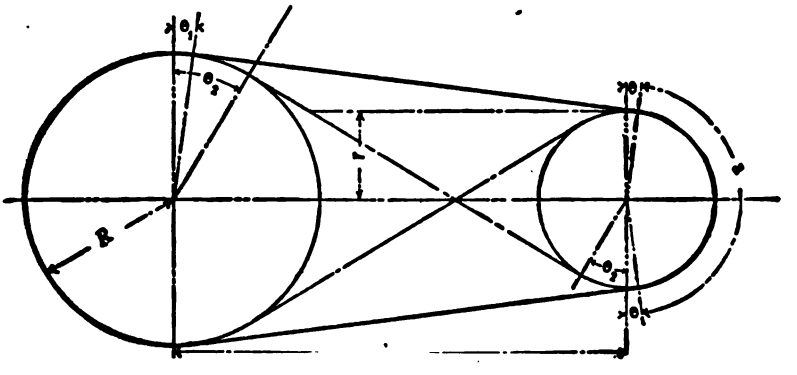


FIG. 178.

From a table of sines find the angle θ in degrees and cos θ.

Then for a crossed belt:

$$L = \left(\frac{\pi}{2} + \theta\right)D + d + 2l\cos\theta \quad . \quad . \quad . \quad . \quad (12)$$

and for an open belt

$$L = \frac{\pi}{2}(D+d) + \theta(D-d) + 2l\cos\theta \quad . \quad . \quad (13)$$

The length of the crossed belt is constant when $D + d$ and l are constant; therefore in designing a pair of cone-pulleys so that the crossed belt will have equal tension on all pairs, it is only necessary to use a pair of equal and similar cones tapering opposite ways.

To design a pair of cone-pulleys for an open belt: Let $D_1D_2D_3D_4$ and $d_1d_2d_3d_4$ = diameters of opposite pulleys (Fig. 179). And using the graphical method given by Mr. C. A. Smith in the A. S. M. E., vol. 10, p. 296, let us suppose the following data to be known:

(1) Diameters of $D_1D_2D_3D_4$ and d_1.

(2) l = distance between centres.

Then let it be required to find the diameters of d_2d_3 and d_4.

C and c are the centres of the opposite cones.

Around centre C draw circles $D_1D_2D_3D_4$ and at centre c draw d_1 to the diameters given.

Draw tangent D_1d_1.

Bisect Cc in the point E and erect a perpendicular EF.

Make the distance $EF = .314l$ found by experiment. With centre F draw arc A tangent to D_1d_1. All lines drawn tangent to arc A will be a common tangent to a pair of cone steps giving the same belt-length as that of the given pair. So to find the diameters of the steps d_2d_3 and d_4 it is only necessary to draw tangents to D_2 and arc A, D_3 and arc A, D_4 and arc A, and with centre c and radii $= cd_2$, cd_3 and cd_4 respectively, draw the circles of the required steps. This method is an approximation, but close enough for all practical purposes.

Exercise 87.—Referring to Fig. 179: First, assume diameters $D_1 = 18''$, $D_2 = 14''$, $D_3 = 10''$ and D_4 and $d_1 = 6''$, and

find the corresponding diameters of the opposite steps according to Smith's graphical method just explained in connection with Fig. 179.

Second, make complete working drawings of one of the cone-pulleys, showing half longitudinal cross-section and half side elevation combined, and also a half end elevation like Fig. 180. *Scale 6″ = 1 foot.*

PROPORTIONS OF CONE PULLEY.

Let t = thickness of edge of rim = a;

h = thickness of hub = $.14\sqrt[3]{BD} + \frac{1}{4}''$ from eq. (10);

H = length of hub = $1.43b$;

R = face radius = $5B$.

The remaining dimensions may be taken from the following table.

TABLE 32.

(Dimensions in inches.)

b	2	2½	3	4	5	6	8	10	12	16	18
a	5/16	5/16	5/16	5/16	7/16	7/16	7/16	1/2	1/2	9/16	9/16
e	1/2	1/2	7/16	5/16	5/8	5/8	5/8	1/2	1/2	1/2	1/2
f	3/4	3/4	7/8	1	1¼	1⅜	1½	1¾	2¼	2¼	2⅝
g	1/2	1/2	5/8	3/4	1	1⅛	1¼	1½	1¾	1¾	1⅞

Rope Pulleys.—Rope pulleys are made of cast iron with grooved rims, as shown in Figs. 181 and 182. The angle of the groove is usually 45°. The grooves for guide pulleys are semicircular at the bottom, the radius of the curve being a little greater than the radius of the rope. The diameter of a

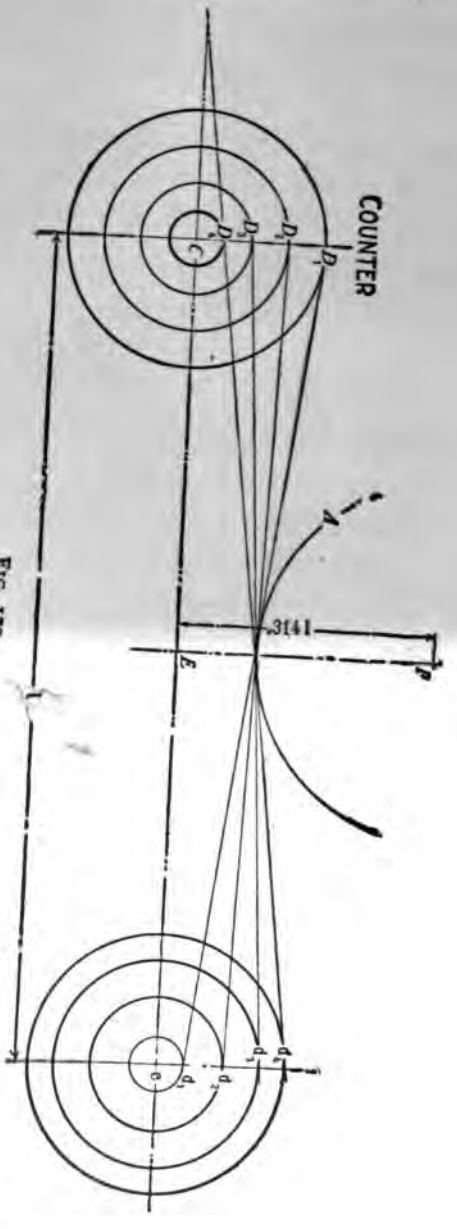


Fig. 179.

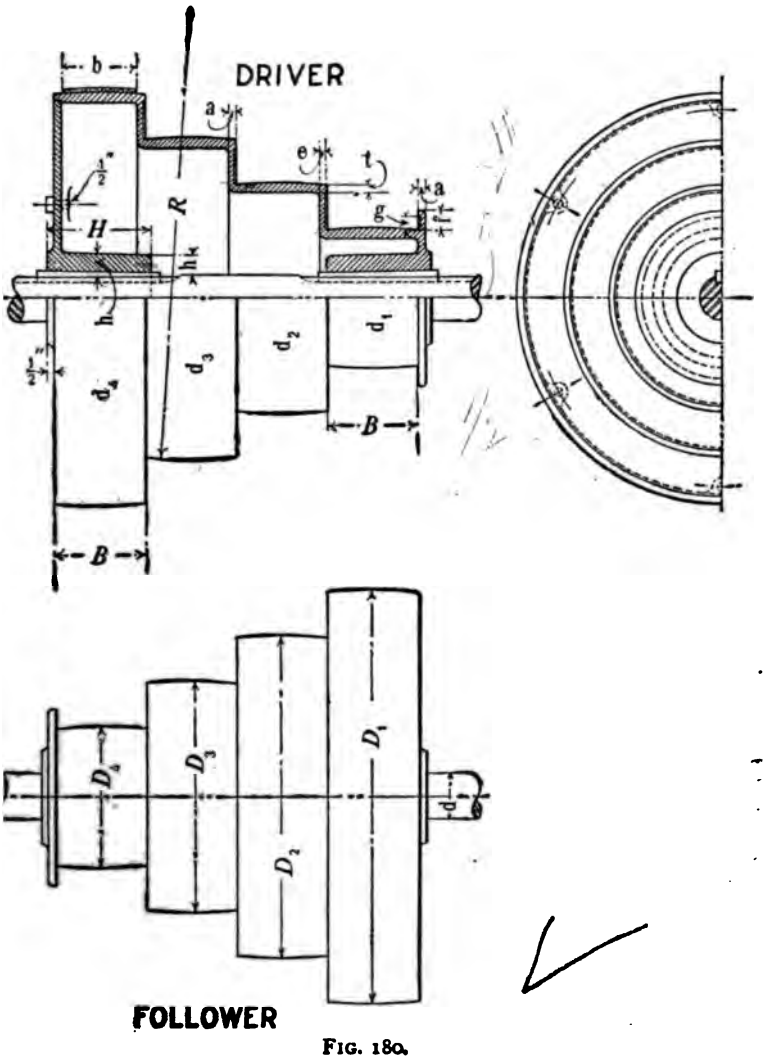


FIG. 180.

rope pulley measured to the centre of the rope should not be less than that given by the following rule:

$$D_1 = (10D + 16)D, \quad \text{where}$$

D_1 = the smallest diameter of the pulley;

D = the diameter of the rope.

As in the case of belt gearing, the slack side of the rope should be on top wherever possible, so as to increase the arc of contact between the rope and the pulley.

Fig. 181. This is the form of groove long used in Great Britain. It has flat sides inclined to each other at from 45° to 60°.

The general practice in America is to use the form of groove shown in Fig. 182, where the sides are curved. This form allows the rope to rotate in the groove, distributing the wear over the entire surface of the rope, making it last longer than it does in the flat-sided groove.

Exercise 88.—Make a drawing of the section of the rim of a rope pulley with five grooves, as shown in Fig. 181. Diam. of rope to be $1\frac{3}{4}''$. *Scale full size.*

Take the other dimensions from the following table.

TABLE 33.

(Dimensions in inches.)

D	A	B	C	E	F	G	H
1	7 10	5 16	$1\frac{1}{2}$	1	13 16	3 4	9/16
$1\frac{1}{4}$	1 2	11 32	$1\frac{13}{16}$	$1\frac{1}{4}$	31/32	15/16	11/16
$1\frac{1}{2}$	9 16	3 8	$2\frac{1}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{8}$	13/16
$1\frac{3}{4}$	5 8	13 32	$2\frac{5}{16}$	$1\frac{3}{4}$	$1\frac{5}{32}$	$1\frac{5}{16}$	15/16
2	11 16	7 16	$2\frac{3}{4}$	2	$1\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$
$2\frac{1}{4}$	3 4	15 32	$3\frac{1}{16}$	$2\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{3}{16}$	$1\frac{3}{16}$
$2\frac{1}{2}$	13 16	1 2	$3\frac{3}{8}$	$2\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{5}{16}$
$2\frac{3}{4}$	7 8	17 32	$3\frac{11}{16}$	$2\frac{3}{4}$	$1\frac{23}{32}$	$2\frac{1}{16}$	$1\frac{7}{16}$

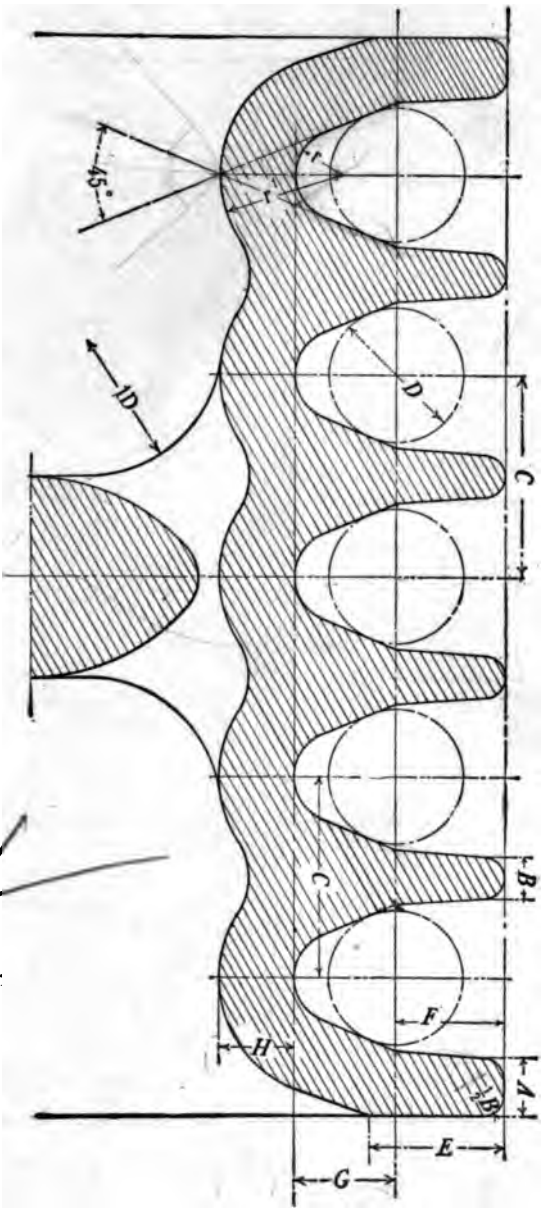
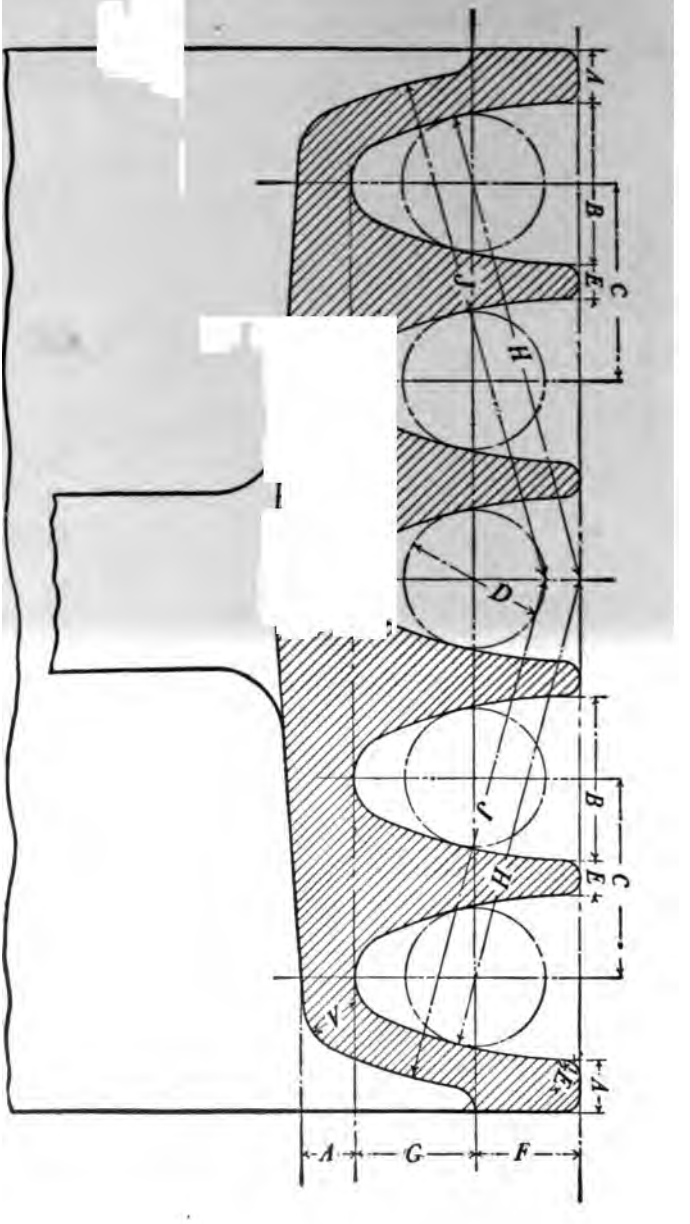


FIG. 181.


A
B
E
C
J
H
D
B
C
J
H
E
A
V
A
G
F

Exercise 89.—Make a drawing of the rope pulley rim section shown in Fig. 182. Diam of rope to be $1\frac{1}{2}$" *Scale full size.* Remaining dimensions may be taken from Table 34.

TABLE 34.

(Dimensions in inches.)

D	*A*	*B*	*C*	*E*	*F*	*G*	*H*	*J*
1	3/8	$1\frac{1}{4}$	$1\frac{1}{2}$	1/4	3/4	7/8	$3\frac{15}{32}$	$3\frac{9}{32}$
$1\frac{1}{4}$	1/2	$1\frac{7}{16}$	$1\frac{3}{4}$	5/16	15/16	$1\frac{1}{8}$	$4\frac{11}{32}$	$4\frac{13}{16}$
$1\frac{1}{2}$	9/16	$1\frac{9}{16}$	$2\frac{1}{8}$	3/8	$1\frac{1}{16}$	$1\frac{5}{16}$	$5\frac{17}{64}$	$5\frac{5}{16}$
$1\frac{3}{4}$	5/8	$2\frac{1}{16}$	$2\frac{1}{8}$	7/16	$1\frac{5}{16}$	$1\frac{1}{2}$	$6\frac{1}{16}$	$6\frac{9}{16}$
2	3/4	$2\frac{3}{16}$	$2\frac{1}{4}$	1/2	$1\frac{1}{8}$	$1\frac{3}{4}$	$6\frac{15}{16}$	$7\frac{1}{8}$
$2\frac{1}{4}$	7/8	$2\frac{1}{16}$	$3\frac{1}{4}$	9/16	$1\frac{11}{16}$	2	$7\frac{51}{64}$	$8\frac{7}{16}$
$2\frac{1}{2}$	15/16	3	$3\frac{3}{8}$	5/8	$1\frac{7}{8}$	$2\frac{3}{16}$	$8\frac{13}{64}$	$9\frac{3}{8}$
$2\frac{3}{4}$	1	$3\frac{5}{16}$	4	11/16	$2\frac{1}{16}$	$2\frac{3}{8}$	$9\frac{17}{32}$	$10\frac{5}{16}$

CHAPTER VIII.

TOOTHED GEARING.

PROPORTIONS OF IRON TEETH. Fig. 183.

p = circular pitch $= k\sqrt{P}$;

p' = diametral pitch $(p \times p') = 3.1416$;

D = pitch diameter $= T \div p'$;

T = number of teeth $= D \times p'$;

l = addendum of tooth $= .3p$;

l' = flank of tooth $= .35p$ to $.4p$;

t = thickness of tooth $= .48p$ for cast-iron teeth,
$= .5p$ for cut teeth;

k = .04 for hand-wheels,
= .05 for ordinary mill gears,
= .06 for wheels of high velocity and mortise gearing;

P = the total force transmitted by one wheel to another through a corner of the tooth $= \frac{550H}{V} = 63020\frac{H}{RN}$;

V = the velocity of the pitch line in feet per second
$= \frac{2 \times 3.1416RN}{12 \times 60} = .00873RN$;

R = the radius of the pitch circle in inches;

N = the number of revolutions of the wheel per minute;

H = the horse-power transmitted by the wheel.

WOOD TEETH or cogs for mortise wheels are usually made nicker than for the iron teeth of the meshing wheel.

t' = thickness of iron teeth to mesh with mortise wheel $= .4p$;

t = thickness of wood cog $= .6p$.

Exercise 90. (Fig. 183.)—*To construct the teeth for a spur gear of* 15 *teeth and rack, p' or diametral pitch* = 2.5. *Involute system, angle of action* = 15°.

Draw the centre line C, and compute the diameter of the pitch circle by dividing the number of teeth by p'.

At the point a where the pitch circle cuts C draw line L, making an angle of 15° with the horizontal pitch line, and draw the base circle tangent to L.

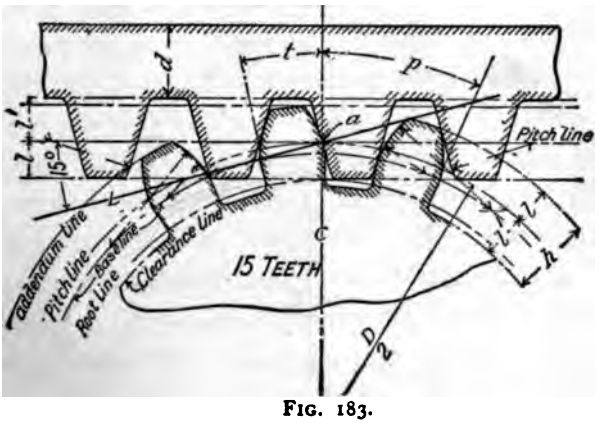


FIG. 183.

To find p: Divide 360° by the number of teeth: the quotient will be the number of degrees in the arc p, which may be laid off with a protractor. Or divide the number of inches in the circumference of the pitch circle by the number of teeth: the quotient will be the pitch. Or divide a quadrant

of the pitch circle with the hair-spring divider into 15 equal parts, and from the point a mark every fourth division for the point where the outline of a tooth intersects the pitch circle. Next lay off the thickness of the tooth equal to half the pitch on the pitch circle of the wheel and the pitch line of the rack.

Draw the addendum line of the wheel with a radius $= \frac{D}{2} + .3p$.

The root line of the drawn tangent to the addendum line of the wheel root line of the wheel is tangent to the addendum the rack.

To describe the involute of the wheel-tooth: Take a piece of tracing-paper or loid, and trace upon it the straight line L, and make a puncture at the point a with a needle. Now at the point where line L is tangent to the base line stick a needle, and rotate line L about it counter-clockwise until it intersects the base line; at the point of intersection stick another needle, and, removing the first needle, adjust the tracing until the line L becomes tangent to the base line at the second needle; then through the puncture a in the tracing, with a 4H pencil sharpened to a conical point mark a point on the drawing-paper: this will be a point on the curve. Continue to find similar points until a sufficient number has been found to form the addendum of the tooth.

It will be seen by the figure that the involute curve forming the addendum of the tooth extends below the pitch line to the base line; this part of the curve is generated in a similar way to the part above the pitch line, except that the generating line L must be rotated in the opposite direction.

The addendum lines of the other teeth may be traced om the one just found.

The rim of the rack, according to Reuleaux, should not e less than δ in thickness, $= .4p + .125$. Unwin gives $.8p$; Low & Bevis give $.47p$. Use Unwin's proportion.

When the curves have been carefully pencilled as above, ıey may be inked in with arcs of circles computed by means the following odontograph table, taken from Geo. B. rant's "Handbook on the Teeth of Gears":

ODONTOGRAPH TABLE—INVOLUTE TEETH.

CORRECTED FOR INTERFERENCE, INTERCHANGEABLE SET.

Teeth.	Divide by the Diametral Pitch.		Multiply by the Circular Pitch.	
	Face Radius.	Flank Radius.	Face Radius.	Flank Radius.
12	2.51	.96	.80	.31
13	2.62	1.09	.83	.34
14	2.72	1.22	.87	.39
15	2.82	1.34	.90	.43
16	2.92	1.46	.93	.47
17	3.02	1.58	.96	.50
18	3.12	1.69	.99	.54
19	3.22	1.79	1.03	.57
20	3.32	1.89	1.06	.60
21	3.41	1.98	1.09	.63
22	3.49	2.06	1.11	.66
23	3.57	2.15	1.13	.69
25	3.71	2.33	1.18	.74
28	3.92	2.59	1.25	.82
31	4.13	2.85	1.31	.91
34	4.33	3.09	1.38	.99
38–40	4.20		1.34	
44–45	4.63		1.48	
52–60	5.74		2.07	
61–70	7.72		2.46	
71–90	9.78		3.11	
91–120	13.38		4.26	
121–180	21.62		6.88	
181–360				

For any intermediate number of teeth proportional intermediate values can easily be found by calculation.

Example.—A gear-wheel has 30 teeth, and the neare number of teeth in the table is 31; then $\frac{5.23 \times 30}{31} = 5.0$ the number to be divided by p' (1.25), making the true fa radius = $4\frac{1}{2}''$ nearly.

The flank of the tooth is radial, and it is joined to the ri with a fillet whose radius is equal to the clearance.

A special rule is provided for the rack-teeth: the fla and one half the face is a straight line drawn at right angl to line L; the other half of the face is a circular-arc cent on the pitch line and a radius found by dividing 2.10″ by p

This rounding of the point of the rack-tooth is necessa when it is to mesh with a pinion having less than *28* teeth.

The following tables will be found convenient for comp ing the diametral pitch with the circular pitch; they from Grant's "Teeth of Gears":

Cir. Pitch.	Diam. Pitch.
6	.52
5½	.58
5	.63
4½	.70
4	.78
3½	.90
3	1.05
2¾	1.15
2½	1.25
2¼	1.40
2	1.57
1¾	1.80
1½	2.10
1¼	2.50
1	3.14
¾	4.20
½	6.28

Diam. Pitch.	Cir. Pitch.
½	6.28
¾	4.20
1	3.14
1¼	2.50
1½	2.10
1¾	1.80
2	1.57
2½	1.25
3	1.05
3½	.90
4	.78
5	.63
6	.52
7	.45
8	.39
9	.35
10	.31

Exercise 91. (Fig. 184.)—*To construct the teeth for a spur-gear wheel and pinion;* wheel to have 40 and the pinion 12 teeth. $p' = 2.10$. Walker system, non-interchangeable.

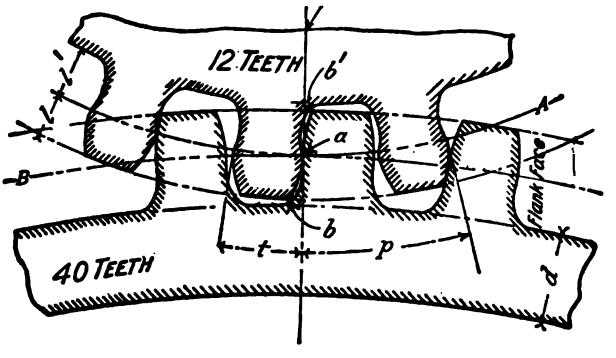


FIG. 184.

The curves of the teeth are epicycloids and epitrochoids, and are found by rolling the pitch circles on each other as follows: For the *addendum* of the wheel-teeth draw arc A on a piece of tracing-paper or celluloid, and place it over the drawing tangent to arc B at the point a. Through the point a on the celluloid make a puncture with a needle, and while holding the needle at a rotate the celluloid a small distance to the right until arc A intersects arc B. At the point of intersection place another needle, and, removing the first needle, adjust the celluloid so as to make arc A tangent to arc B at the second needle, and through the puncture mark a point with the pencil; this will be a point in the curve of the face edge. Other points may be found in a similar way to complete the curves required.

For the face edge of the pinion-tooth roll arc B on arc A, and the point a will describe the curve ab.

To draw the *flank* of the wheel-tooth: When arc *A* on the celluloid is tangent to arc *B* at *a*, trace curve *ab* on the celluloid and make a puncture through *b*; then roll arc *A* to

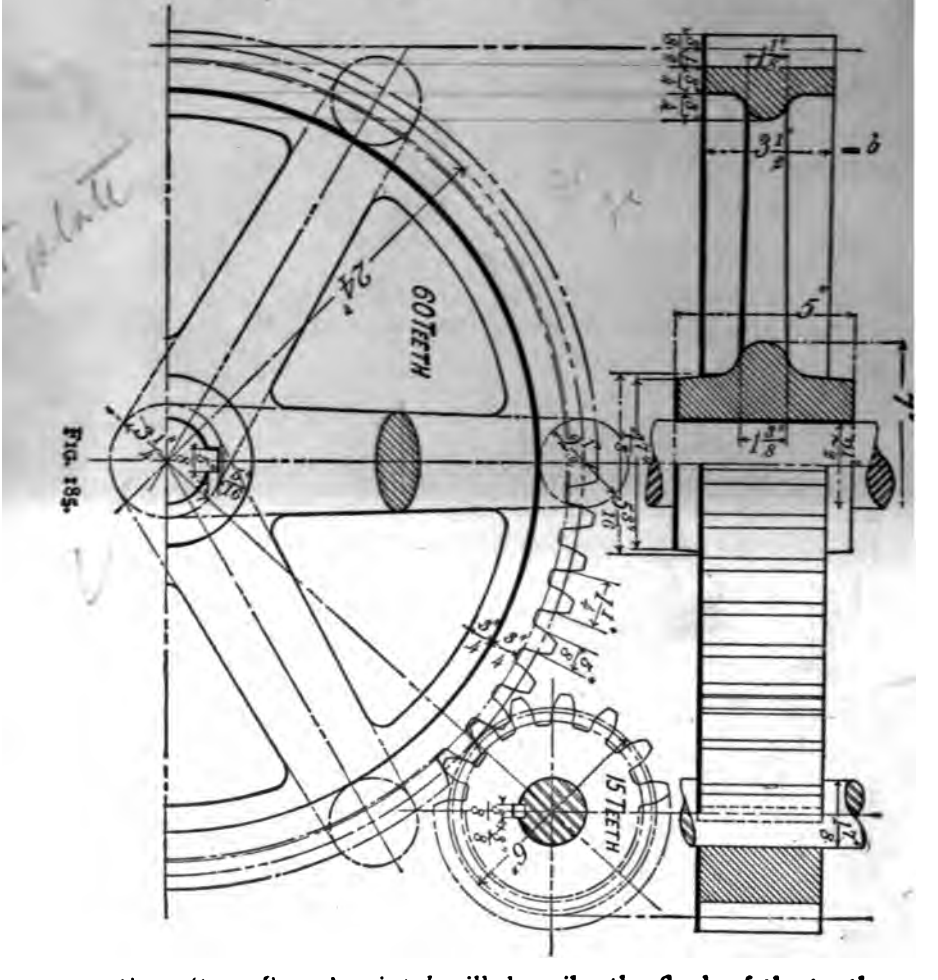


FIG. 185.

the left on *B*, and point *b* will describe **the flank of the tooth.** The flank of the pinion-tooth is then found **by rolling arc *B*. on arc *A*, when the point *b'* will describe the curve.**

Exercise 92. (Fig. 185.)—*Draw the* HALF ELEVATION, HALF PLAN, *and* HALF SECTIONAL PLAN *of a spur-gear wheel and pinion;* the wheel to have 60 and the pinion 15 teeth. $p' = 2.5$.

Draw all the teeth in one quadrant of the elevation, involute system. Fig 185 is the drawing of a spur-gear wheel made by Messrs. Robert Poole & Sons of Baltimore, Md., and presented to Sibley College for use as a model in the drafting-room.

Exercise 93. (Fig. 186.)—*Draw* ELEVATION, CROSS-SECTION, *and* PLAN *of a bevel-gear wheel and pinion.* The axes are to be at right angles to each other, and the wheel is to have 50 and the pinion 24 teeth. $p' = 2.10$. Radial flank system, non-interchangeable.

Draw centre lines C and C' at right angles to each other, find the radii of the pitch circles, and draw D and D' at the proper distance from the axes. Draw E and E' at right angles to each other. F and F' are the developed pitch circles on which the teeth are drawn, the same as if they were for spur gears. And since the flanks are radial, the rolling circles A and B used to generate the face curves of the teeth are equal in diameter to the radius R and R' of the developed pitch circles of the pinion and wheel respectively.

A model of this wheel will be found in the drafting-room for use in connection with this problem.

Exercise 94. (Fig. 187.) — *Construct a worm-wheel and worm;* the wheel to have 50 teeth, and the worm-teeth to be drawn like those of the involute rack; that is, the face edge will be drawn at right angles to line L, when line L makes

the angle of 15° with the horizontal pitch line *H*, as shown by the longitudinal cross-section in Fig. 187.

The teeth of the wheel are made by a cutter similar to

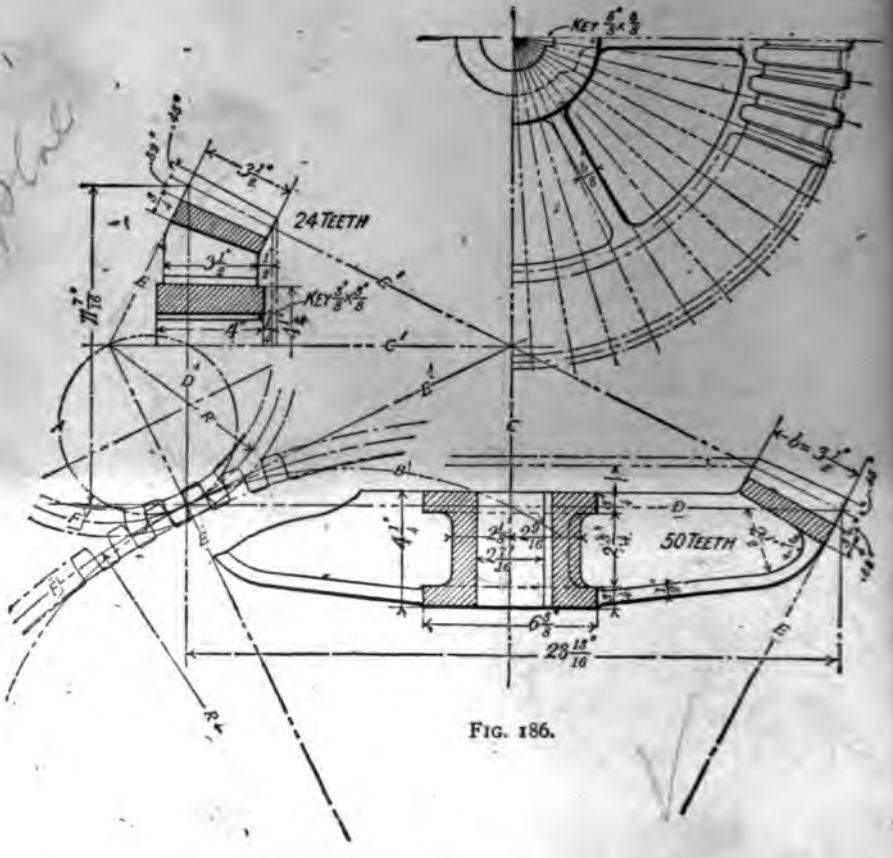


FIG. 186.

the worm, except that grooves are cut in the threads parallel to the axis, and the material is hardened steel. The worm itself is usually made of cast iron, but is sometimes made of wrought iron or malleable cast iron.

The horizontal pitch line should be so placed as to bisect the cross-sectional area of the wheel-tooth at a; otherwise the proportions of the teeth may be the same as those used for wheel and rack.

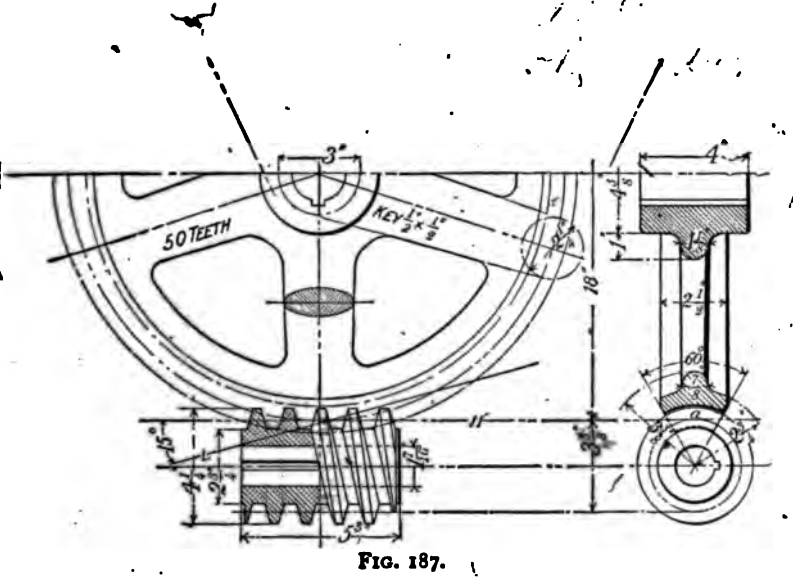


FIG. 187.

Exercise 95. (Fig. 188.) — *Design a cast-iron gear-wheel given the pitch-circle diameter* 51", *revolutions per minute* 90, *horse-power transmitted* 280.

First find the whole pressure of one wheel on the other $= P = \frac{550H}{V}$; ($V = .00873RN = .00873 \times 25.5 \times 90$;) then find the circular pitch $p = .0447\sqrt{P}$.

The number of teeth can now be found by multiplying the diameter of the pitch circle $D \times 3.1416$, and dividing by $p = \frac{51 \times 3.1416}{p} =$ the nearest *even* number.

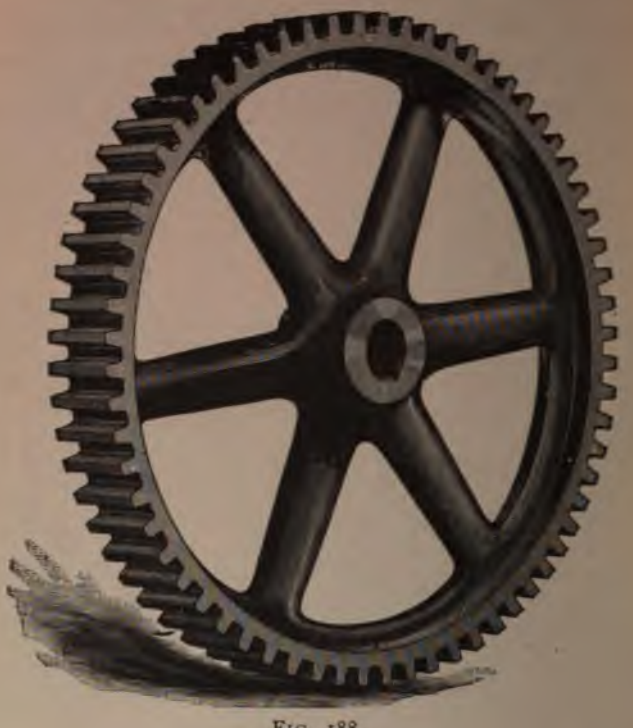


FIG. 188.

Let T represent the number of teeth; then the velocity of the pitch line may be expressed as follows:

$$V = \frac{\pi TN}{12 \times 16},$$

and the pressure on the teeth is

$$P = \frac{550 \times 12 \times 60 \times H}{\pi TN} = 396000 \frac{H}{\pi TN}.$$

Taking the width of the teeth into consideration, let

$t = .36p$ for iron teeth when worn,

$= .45p$ for wood teeth when worn;

$h = .7p$ for iron teeth,

$= .6p$ for wood teeth; then

$$P = .046bpf \text{ for iron teeth,}$$
$$= .084bpf \text{ for wood teeth;}$$

and $p = k_1\sqrt{\frac{p}{b}}\sqrt{P}$ when b = width of tooth = from 2 to $4p$, and in practice

$$k_1 = .0707 \text{ for iron wheels,}$$
$$= .0848 \text{ for mortise wheels.}$$

When $b = 2.5p$, Unwin gives $p = .0447\sqrt{P}$

Low & Bevis give $p = \sqrt{\frac{P}{200T}}$.

The dimensions of the teeth may be determined from the proportions already given: b = the breadth of face $= 2.5p$, etc.

As the shaft for this wheel would probably have to resist a combined twisting and bending action, we can assume the diameter of the shaft to be 6″, and the wheel fit 7″.

The *width* and *breadth* of the arms, the *thickness* of the rim, and the *thickness* and *length* of the hub, etc., can be easily determined by the proportions given in the following pages.

Arm of Gear Wheels.—The usual shapes of *arm cross sections* are shown in Figs. 189 to 192. Fig. 189 is mostly used for pulleys and light wheels; Fig. 191 shows another section that is commonly used in light spur wheels, that in Fig. 192 for heavy spur gears, and that in Fig. 189 for bevel gears.

FIG. 189.

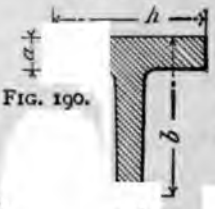


FIG. 190.



FIG. 191.

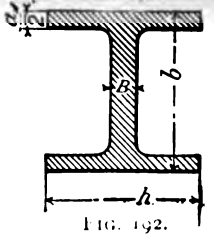


FIG. 192.

When $a = .48p$ = the thickness of [illegible] Unwin gives $h = \frac{.758}{\sqrt{n}}\sqrt{bR}$, m[illegible] the centre of the wheel. T[illegible] 12″ on each side toward [illegible] = the number of arms; R = [illegible] the wheel; b = the width of [illegible] feathers, which may be = the [illegible] of the teeth as shown at b in Fig. 193, or $\frac{5}{4}$ the breadth of the teeth measured at the centre of the shaft and from $\frac{8}{4}$ to $\frac{15}{16}$ at the rim.

The ribs or feathers B do not add much to the resistance of the arms to bending in the direction of the driving force, but they are necessary to give lateral stiffness to the arms. Unwin gives $B = .3p$. The feathers should be tapered to facilitate the removal of the pattern from the sand.

To determine the number of arms in a wheel, Low & Bevis give $\frac{D}{36} + 4$. The nearest number divisible by 2 should be taken.

Unwin gives four arms for wheels not over 4 ft. in diameter, six arms for wheels of from 4 to 8 ft. in diameter, and eight arms for wheels from 8 to 16 ft. in diameter.

Rims of Gear Wheels.—The usual rim sections are shown in Figs. 193 to 204. The section shown in Fig. 193 is commonly used in light wheels.

The following proportions agree closely with most authorities on the subject: d = the thickness of the rim at the edge = $.48p$. The other proportions are shown in the figures.

In the rims for bevel gears shown in Figs. 198 to 200 the thickest part of the rim should be $\frac{6}{5}d$.

Figs. 201 and 202 show examples of mortise gears for

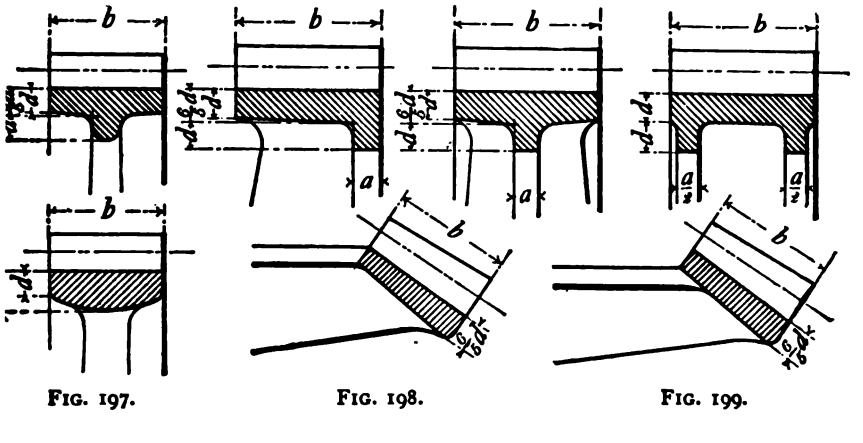


FIG. 193. FIG. 194. FIG. 195. FIG. 196.

FIG. 197. FIG. 198. FIG. 199.

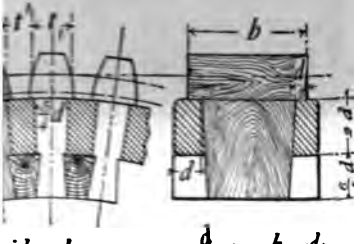
spur and bevel wheels respectively; the mortise teeth are fixed either by wood keys as shown in Fig. 201, or by round iron pins as shown in Fig. 202. The proportions given in the figures agree closely with good practice.

Shr g.—When the rim of a wheel is wider than the teeth and extends towards the point so as to form an annular ring uniting the ends of the teeth, the teeth are said to be shrouded. Figs. 203 and 204 give two examples of shrouded teeth. By shrouding out to the pitch circle as shown in Fig. 203, teeth which are no thicker at the root than at the pitch circle can be strengthened about 100 per cent. In the pinion of a pair of gear wheels the shrouding may extend to the points of the teeth in Fig. 204; this compensates for the weak form h in very small wheels, and prevents their failure from ve wear.

FIG. 200.



FIG. 201.



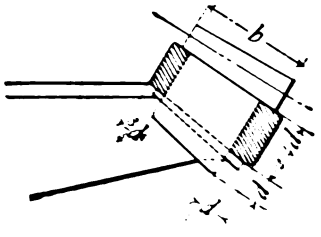


FIG. 202.

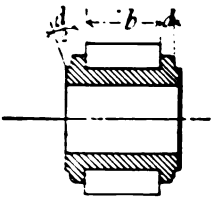


FIG. 203

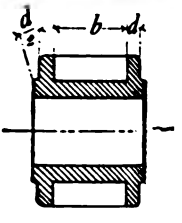


FIG. 204.

Hubs of Gear Wheels.—Figs. 205, 206, and 207 give examples of hubs to correspond to the examples of arms shown in Figs. 189, 191, and 192, respectively.

The thickness of metal surrounding the bore of a gear

wheel is given by Reuleaux = $w = .4h + .4''$ (when h = the width of the arm measured at the centre of the wheel). The keyway should be cut the full length of the hub, and the metal reinforced over the keyway if the wheel is intended for heavy duty. In large wheels the hubs are sometimes strengthened by wrought-iron rings shrunk on both ends; the thickness is made $= \frac{w}{2}$, and the thickness of the metal under the rings is $\frac{3}{4}w$. b = width of teeth.

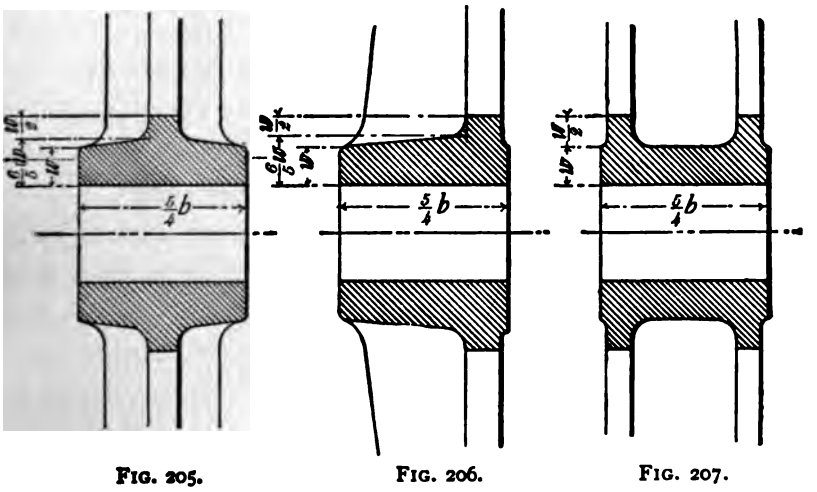


FIG. 205. FIG. 206. FIG. 207.

In heavy wheels with a large amount of metal surrounding the bore, the hub is sometimes slotted across between the arms to give relief from initial strains due to unequal contraction in cooling; these slots are then filled with metal strips, and the divided hub is held firmly together by the iron or steel ring referred to above.

CHAPTER IX.

VALVES, COCKS, AND OIL-CUPS.

Valves.—A for regulating the flow of a fluid through an o

Prof. Unwin di ito three classes:—(1) Flap-valves, or those wh a hinge; (2) lift-valves, or those which rise pe to the seat; (3) slide-valves, or those which move eat. The valve-face is that part of the valve in its seat when closed.

Foot-valve and Str Foot-valves are used to hold the water in long suction-pipes; otherwise the pump would have to be charged every time before starting.

The strainer protects the valve from being choked with stones or other solids. The most common foot-valves are made of two cast-iron boxes, called the valve-box and strainer, bolted together by flanges, and having a leather clack-valve between them. The lower box is perforated with circular holes ⅛ to ¼ diameter, and is called the strainer or snore-piece. In small foot-valves the suction is generally screwed into the top of the valve-box.

Fig. 207 shows a vertical section and three half plans of a foot-valve for a 6 suction-pipe. *VB* is the valve-box, *S* the strainer, *A* is the valve-seat, *B* main valve, and *C* an auxiliary

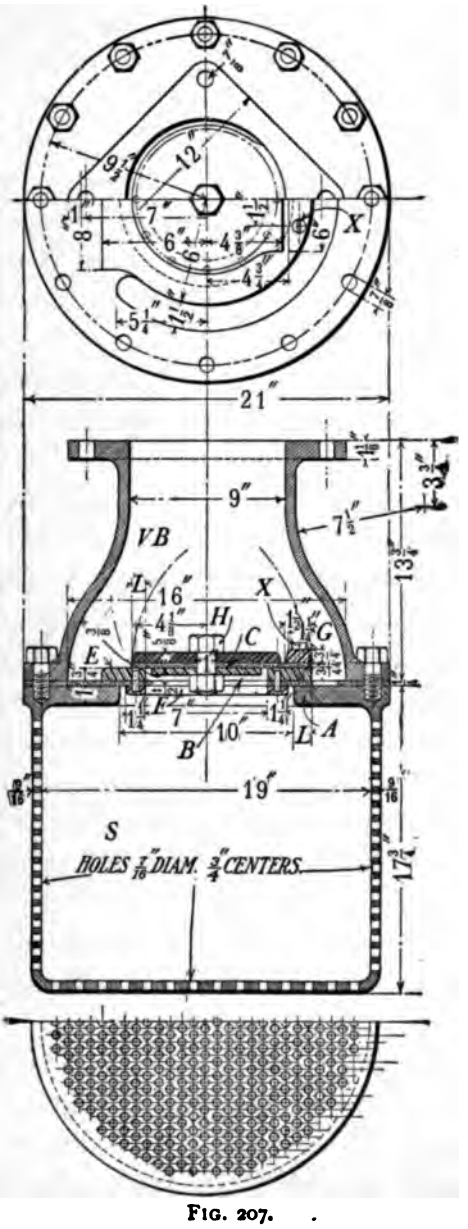


FIG. 207.

valve on top of *B*. This style of clack is called a *relief* or *break* clack. Mr. Henry Teague, of Lincoln, England, in a paper read before the Inst. of M. E. of England, in 1887, reported having used a 15″ main clack with a 5″ supplementary clack for the purpose of reducing the very great concussion which was had by using the 15″ clack alone, with the result that even when the hand or the ear was placed on the clack-box hardly a tremor or a sound was perceptible. *D* is the entrance to the

This double-valve s almost complete freedom from shocks even in , and therefore works very quietly.

The main valve, ma leather, forms the joint between the valve-box a ier. *E* is the top and *F* is the bottom valve-plate, t gether with $\frac{5}{8}$-inch rivets, and an opening in the centre al to an area of about one half or one third that of the main opening. This auxiliary opening is fitted with the clack-valve *C* referred to above. It has an upper and a lower valve-plate, held together with the bolt *H* and fastened to the main valve with two screws at *X*, in plan and sectional elevation.

The laps *L* should be made one tenth of the diameter of the respective valve-openings.

Exercise 96.—Make drawings of foot-valve and strainer shown in Fig. 207, and also an outside elevation of the valve-box and strainer. *Scale 3″ = 1 foot.*

India-rubber Valve.—This valve (Fig. 208) consists of an india-rubber disk *D*, a brass grating or seat *s*, and a perforated brass guard. The rubber guard and valve are attached to the grating by a stud-bolt *B*. The purpose of the guard

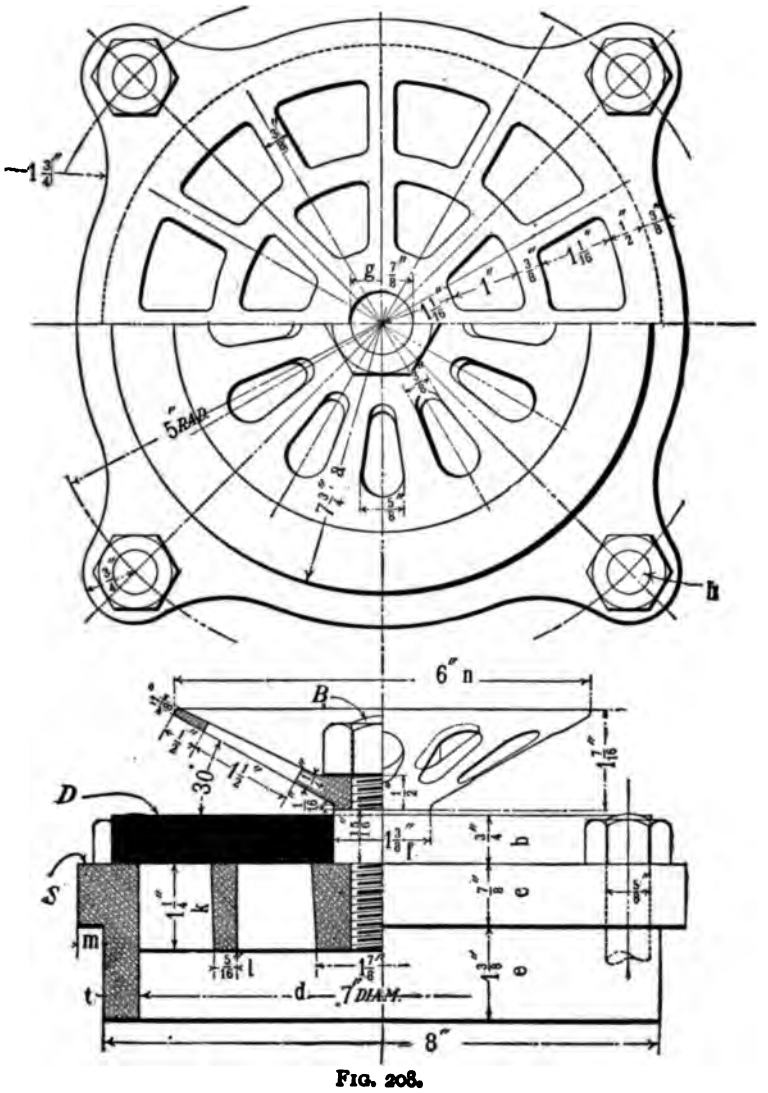


FIG. 208.

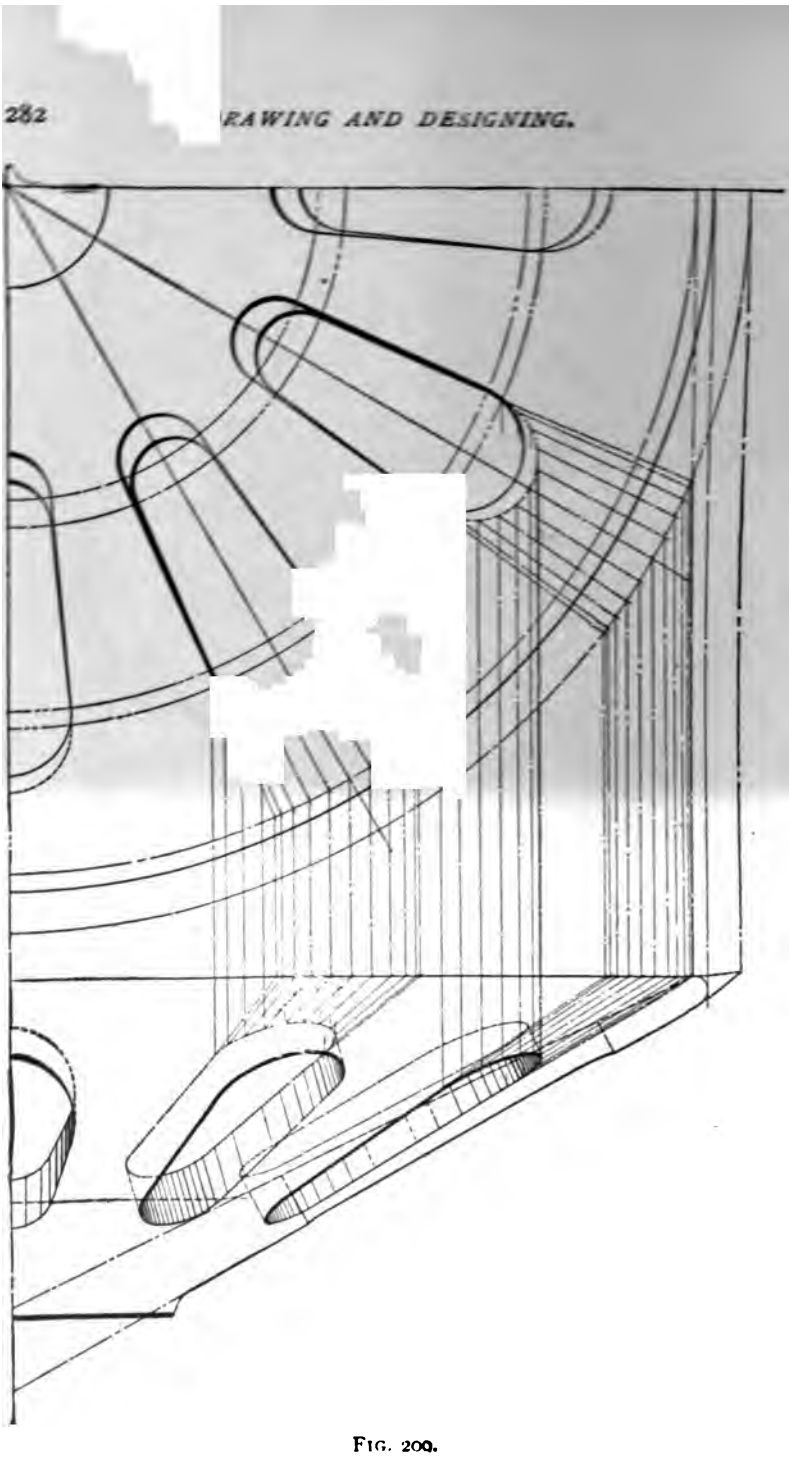


FIG. 200.

is to prevent the valve from rising too high. The perforations in the grating should not be large enough to cause much flexure of the rubber disk. The area of the grating should be such that when the valve is closed the pressure does not exceed 40 lbs. per square inch.

The thickness of the india-rubber disk for large valves—i.e., valves over 6″ in diameter—in condensers and pumps should be $\frac{3}{4}$″ to $\frac{7}{8}$″. India-rubber valves are not good for pressures over 100 lbs. per square inch.

Exercise 97.—Make a complete drawing of the india-rubber valve as shown in Fig. 208. *Scale full size.* The projection of the perforations in the conical guard is shown in Fig. 209.

The following proportions represent good practice. Use the nearest $\frac{1}{16}$″. Unit = $.19\sqrt{d}$.

a = diameter of india-rubber disk	= 15.5	of unit.
b = thickness of the india-rubber disk	= 1.6	"
c = thickness of the grating-lip	= 1.75	"
d = diameter of the valve.		
e = depth of seat-body	= 2.75	"
f = diameter of stud-body	= 2.75	"
g = diameter of stud	= 1.75	"
h = diameter of holding-down bolt	= 1.25	"
k = depth of grating	= 2.50	"
l = thickness of grating-rib	= .65	"
m = width of seat-lip	= .75	"
n = diameter of guard	= 12.00	"

Exercise 98.—Make a complete drawing of an india-rubber disk-valve similar to Fig. 208. $d = 10$″. *Scale 9″ = 1 foot.*

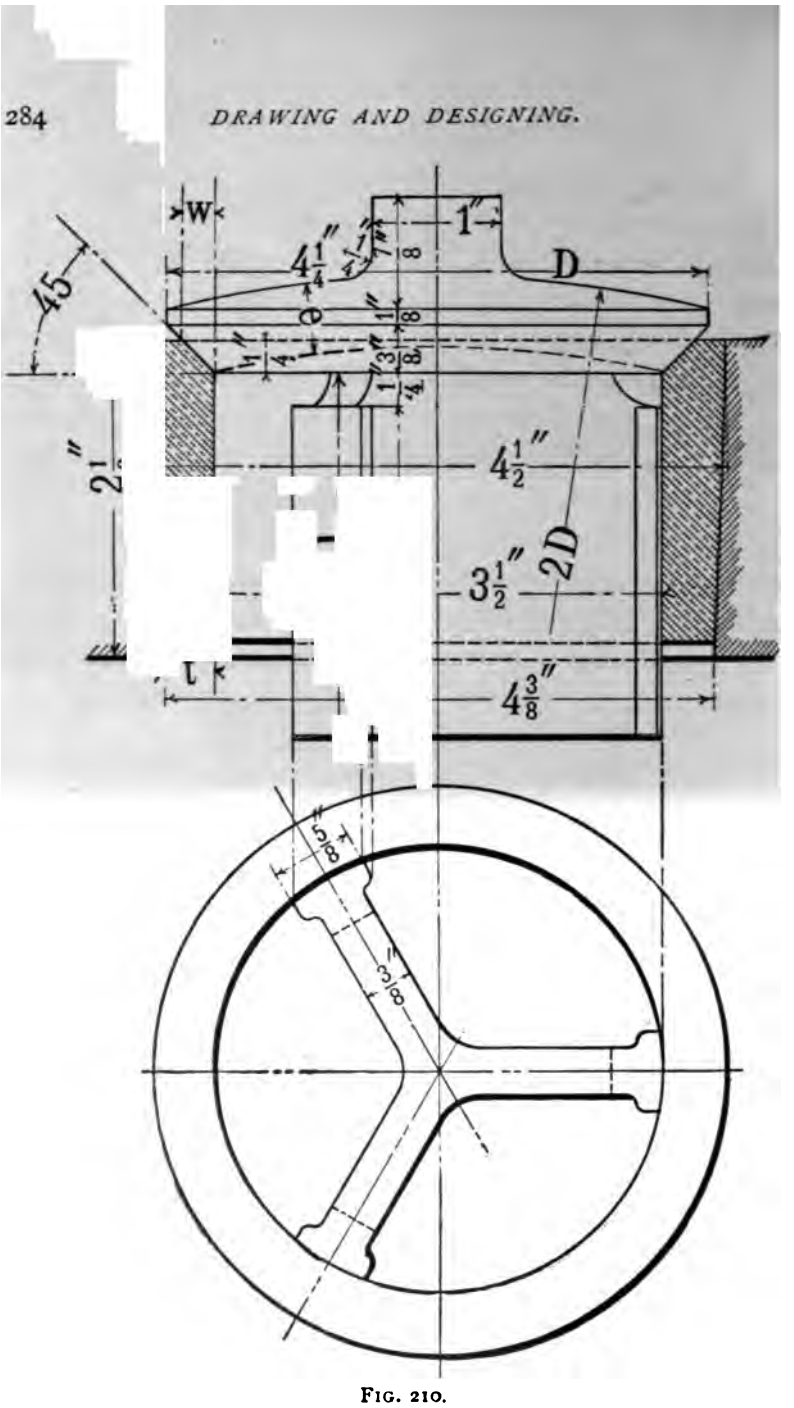

W
1″
D
45
$4\frac{1}{4}$″
$\frac{1}{8}$″
$\frac{1}{4}$″
$\frac{3}{8}$″
$\frac{1}{4}$″
$2\frac{1}{2}$″
$4\frac{1}{2}$″
$3\frac{1}{2}$″
2D
L
$4\frac{3}{8}$″

FIG. 210.

Lift- or Wing-valves (Fig. 210).—These valves are usually made of brass. The essential features are a circular disk and seat. The edges between the disk and seat are bevelled to the angle of 45°, and are easily fitted and ground together. Springs or rods are used to close these valves when it is necessary to place them in a horizontal position. To give the valve a partial rotation and provide a new seating at each stroke the wings are curved slightly, as shown at Fig. 211. The curving is arbitrary, and may be projected as shown in the figure. The outside of the seat has usually a taper of $\frac{1}{8}''$ in $12''$, but is sometimes driven straight. The amount of the lift of the valve may be determined as follows:

Let

a = area of opening in seat;
d = diameter of opening in seat;
L = lift of valve.

Then

$$a = .7854d^2 \text{ and } L = .35d. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Taking a unit of proportion $= .2\sqrt{d}$, then

t = thickness of disk = 1.3;
l = length of wings = 8;
t = thickness of seat = 1 at small end.

Exercise 99.—Draw the valve as shown in Fig. 210 to the dimensions given. *Scale full size.*

Exercise 100.—Make drawing of the curved wing-valve as shown in Fig. 211. *Scale full size.*

Spindle-valves (Fig. 212).—These valves are guided centrally by means of a spindle and bridge; otherwise they are

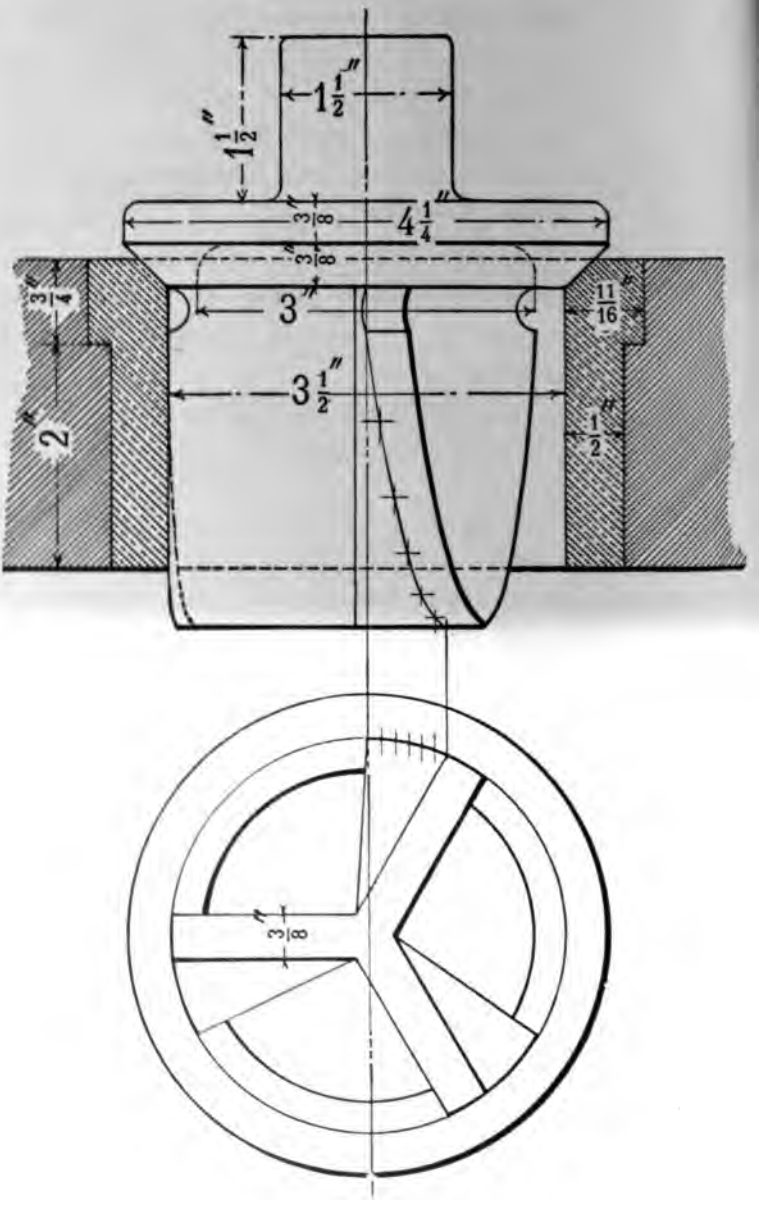


FIG. 211.

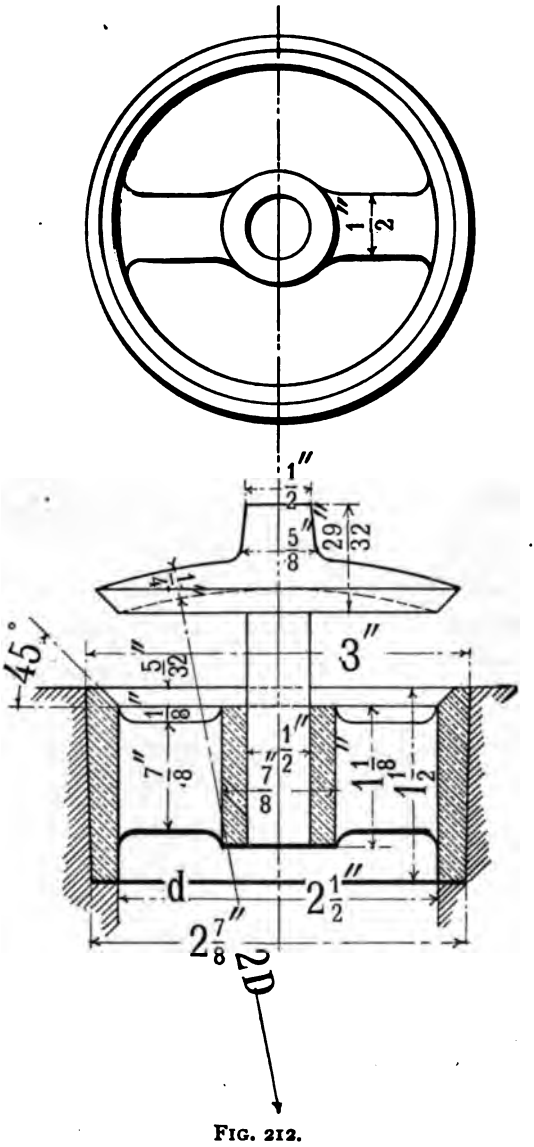


FIG. 212.

similar to the wing-valve, but used for light work in pumps. The wing-valve and the spindle-valve are sometimes made with a flat seat and a leather face and also used for light duty in pumps, but have no advantage over the bevelled metal edges.

Let W (Fig. 210) = the width of the bearing-edges measured perpendicularly to the axis of the valve, p = the maximum difference of pressure on the two sides of the valve; then $\frac{p}{\pi d W}$ = the crushing pressure per square inch on the narrow bevelled edges of the valve and seat.

The greatest safe pressure per square inch for phosphor-bronze is 3000 lbs.; for gun-metal, 2000 lbs.; cast iron, 1000 lbs.; and leather and india-rubber, 700 lbs.

Exercise 101.—Make drawings of the spindle and valve as shown in Fig. 212. *Scale full size.*

Ball-valves (Fig. 213).—These valves are much used in deep well-pumps and small fast-running pumps. To guide the lift of the ball it is surrounded by a cage with three or four ribs. The ribs should be as narrow as safety will permit, so as not to interfere with the free flow of the fluid above the valve-seat. Gun-metal is the best material for the balls. To lighten them they should be made hollow.

The usual proportions for the ball-valves are given below:

$$\text{Unit} = .2\sqrt{d}.$$

a = diameter of ball	$= 1.34d$.
b = inside diameter of seat-casing	$= 1.12d$.
c = thickness of ball-guide	$= .9$ times unit.
e = distance between guides	$= a + \frac{1}{16}''$.

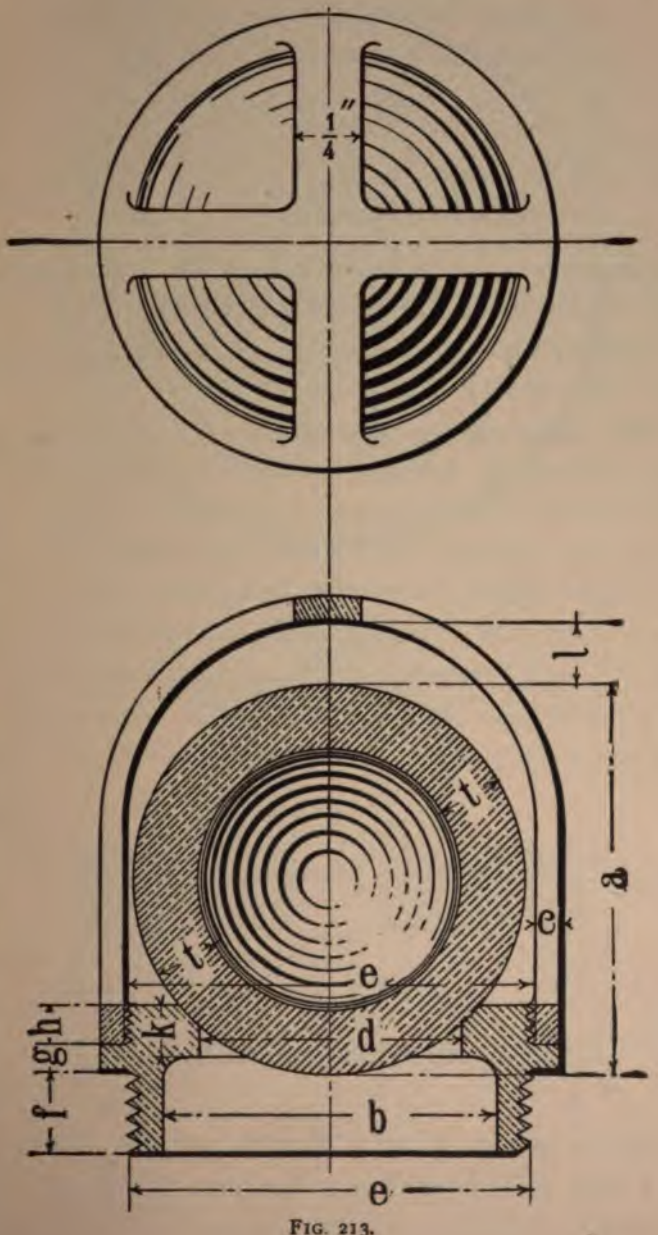


FIG. 213.

f = length of seat-shank	= 3 times unit.
g = thickness of seat-flange	= 1 "
h	= 1.2 "
k	= 1.8 "
l = lift of valve	= .16d "
t = thickness of ball-shell	= 1.2 "

These valves work best with a small lift. William M. Barr says that the lift of ball-valves should not exceed $\frac{1}{4}''$.

Exercise 102.—Make drawings similar to those shown in Fig. 213. $d = 1\frac{1}{2}''$. *Scale $1\frac{1}{2}$ full size.*

Flat India-rubber Disk-valves.—Fig. 214 shows an ordinary example of this style of valve for cold water. The valve-seat and spindle are cast in one piece. The spindle is turned and polished, and the hole in the india-rubber disk is $\frac{1}{16}''$ larger than the diameter of the spindle. This allows free action of the valve. The valve-seat is screwed into place with a pitch of eight threads to the inch, which may be maintained for all sizes up to $4\frac{1}{2}''$ diameter. Mr. W. M. Barr gives the following dimensions for india-rubber valves:

TABLE 35

Diameter.	Thickness.	Hole.
2″	3/32″	1/2″
2½″	7/64″	5/8″
3″	1/8″	9/16″
3½″	5/64″	3/4″
4″	5/64″	11/16″
4½″	3/32″	13/16″
5″	7/64″	15/16″

Springs give good results if made with No. 12 brass wire for 2″ and $2\frac{1}{2}''$ valves; No. 10 wire for 3″ and $3\frac{1}{2}''$ valves.

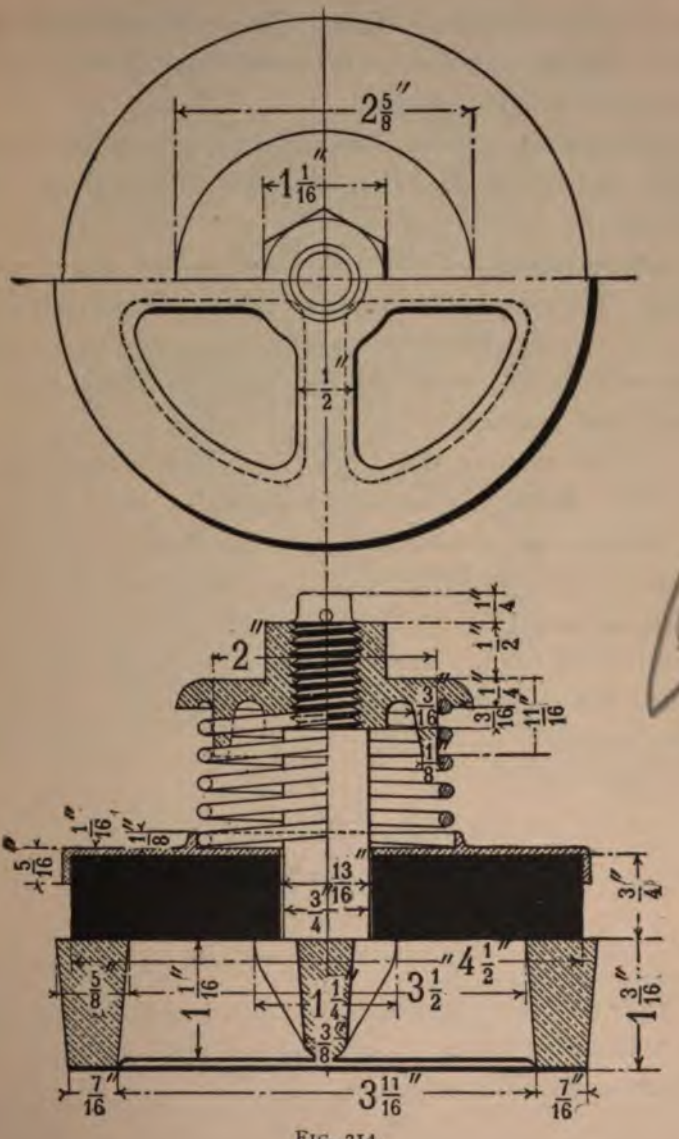

2 5/8"
1 1/16"
1/2"
2"
1 1/4"
1 1/2"
3/16"
1 1/4"
11/16"
1/8"
1 1/16"
1 1/8"
5/16"
13/16"
3/4"
3/4"
1 1/16"
1 1/4"
3 1/2"
4 1/2"
3/8"
1 3/16"
7/16"
3 11/16"
7/16"

FIG. 214.

and No. 8 for 4" and 4½ valves. The outside diameter of the spring may be = .5 that of the valve-disk. Five to six coils will give a suitable elasticity.

Exercise 103.—Make drawings for the india-rubber flat-disk -valve, as shown in Fig. 214, to the dimensions given. *Scale full size.*

Globe-valves.—These valves are opened and closed by hand. The valve in Fig. 214 is for steam. When such a valve is used for cold water the valve-face is made of leather or india-rubber, and when for hot water the india-rubber is mixed with graphite.

The construction of the valve is so plainly shown in Fig. 215 that a description seems unnecessary.

Exercise 104.—Make drawings of globe-valve as shown in Fig. 215, and also a right-end elevation. *Scale full size.*

Stop-valve (Fig. 216).—This is another style of lift-valve controlled by hand. The particular valve shown in the figure is used as a throttle-valve by the Ball Engine Co., who kindly sent drawings.

Let

t = thickness of casing;
p = pressure in lbs. per square inch;
d = diameter of the sphere in inches;
f = safe bursting strength of material.

Take 2000 for cast iron and 17,500 for yellow brass, and use a factor of safety 8, which gives 2500 for the former and about 2200 for the latter; then

$$t = \frac{pd}{4f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

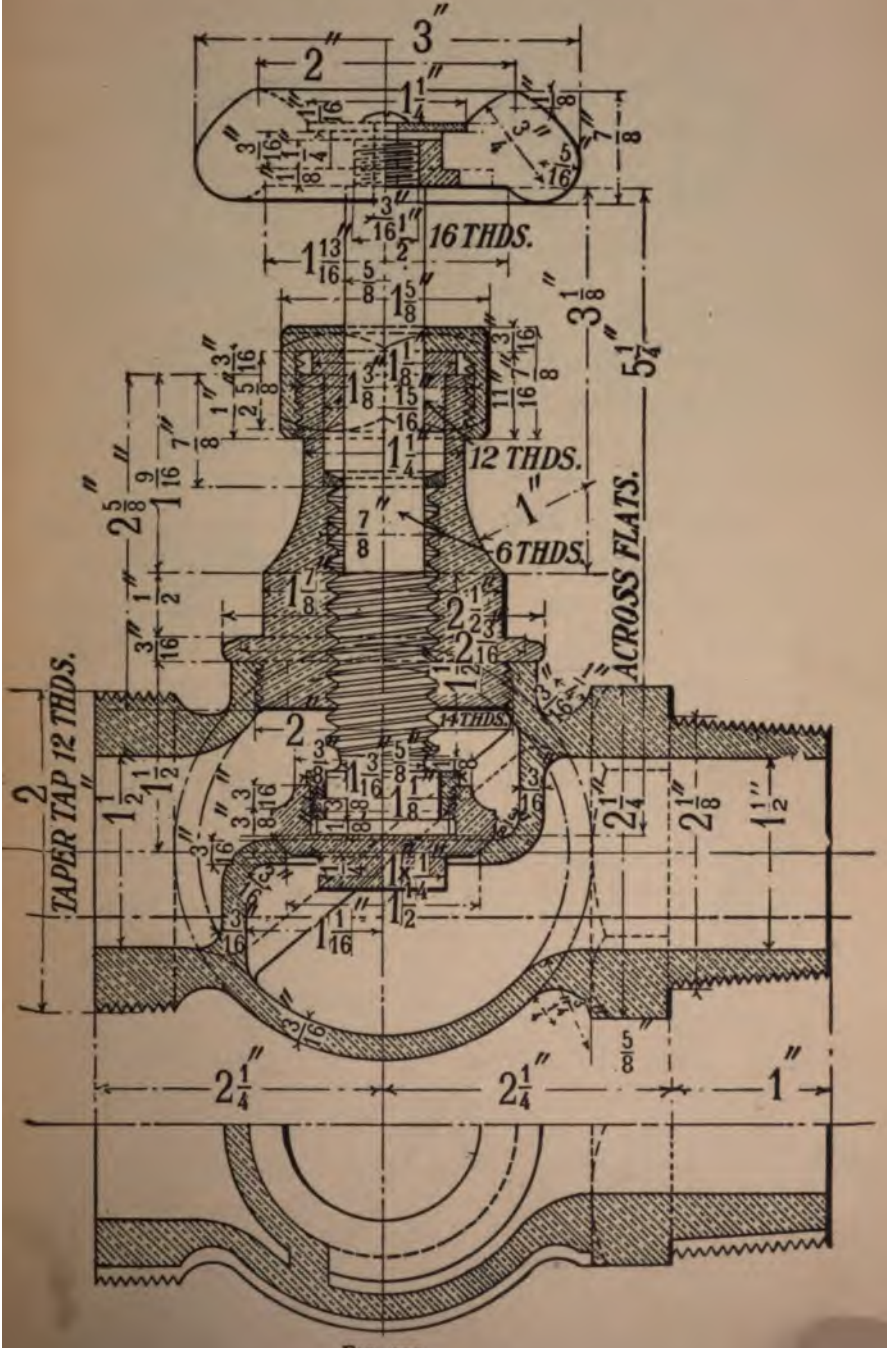


FIG. 215.

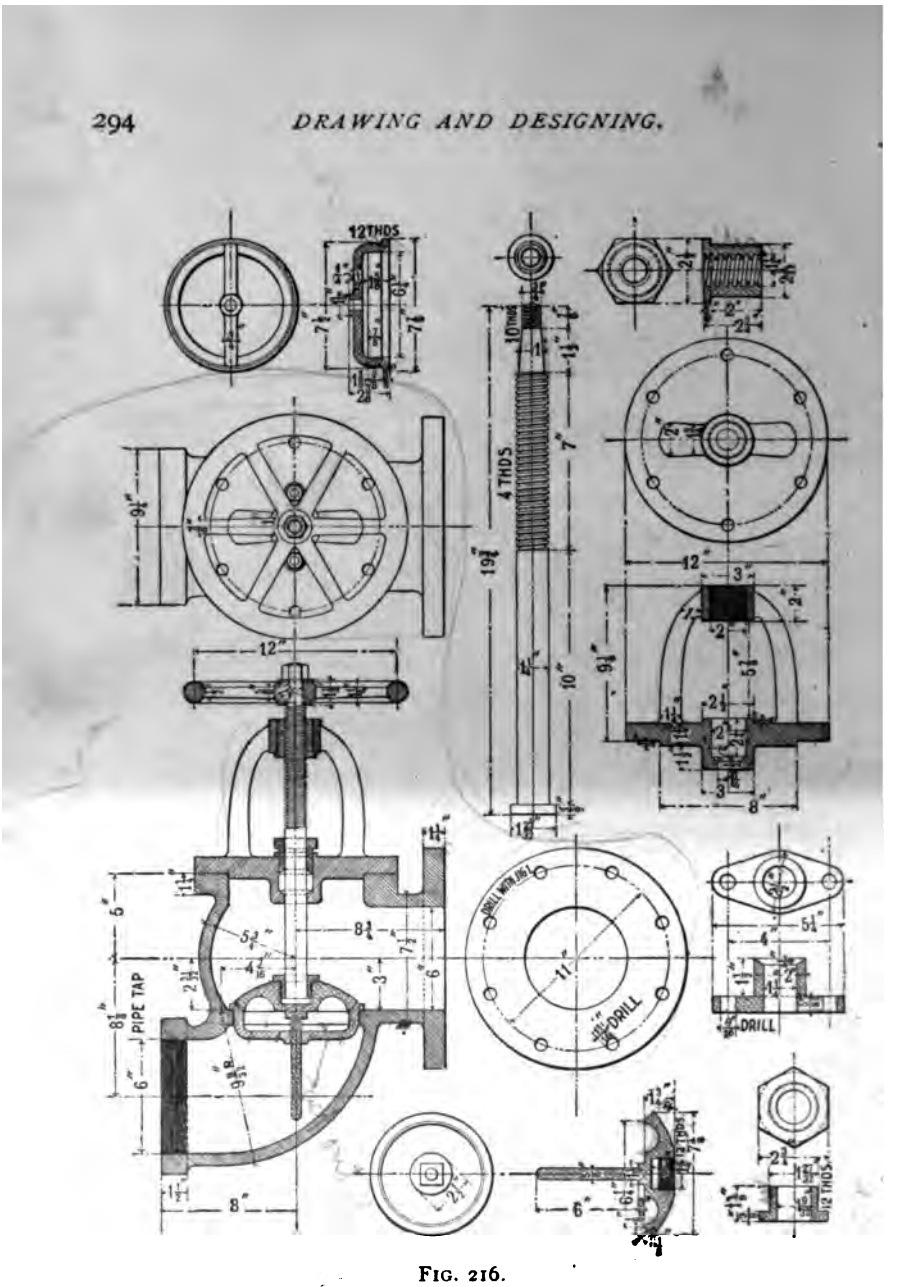


FIG. 216.

The lift of the valve may be determined by formula (1) or winged lift-valves. The valve and its seat must pass hrough the valve-chest, so the opening should be made about ;" larger than the outside diameter of the valve-seat.

The length of the thread on the valve-stem is equal to :he length of the nut + lift of valve + $\frac{1}{4}$" for clearance.

Exercise 105.—Make drawings of the stop-valve as shown n Fig. 216. *Scale 4" = 1 foot.*

Make the diameter of the inlet $6\frac{3}{4}$" to the root of the hread, instead of 6" as shown in the figure.

Boiler Check-valve.—Fig. 217 shows working drawings of ie Foster Safety Boiler-check.

Exercise 106.—Make drawing of the Foster Boiler-check s shown in Fig. 217. Scale, full size.

Cocks.—Cocks are valves which operate with a rotary otion. The most common style of cock is that which con-sts of a plug made in the form of a truncated cone rotating a a seat of the same shape cast on a pipe.

In Fig. 218 *P* is the plug, and *C* the casing or conical seat.) is the opening through the plug. By rotating the plug in ne direction the openings are brought in line with the inlet ! and outlet *B* of the pipe or casing. In this position the ock is open. Further rotation through 90° in either direction ill bring the openings in the plug opposite the solid parts of ie casing and close the valve.

Exercise 107.—Make drawings of the *blow-off* cock shown i Fig. 218, and in addition to the views given make a *half ctional plan* and *half sectional end view.* Scale, full size.

In Fig. 219 is shown a blow-off cock which is really a *ing-valve*, opened and closed by a piston which in turn is op-

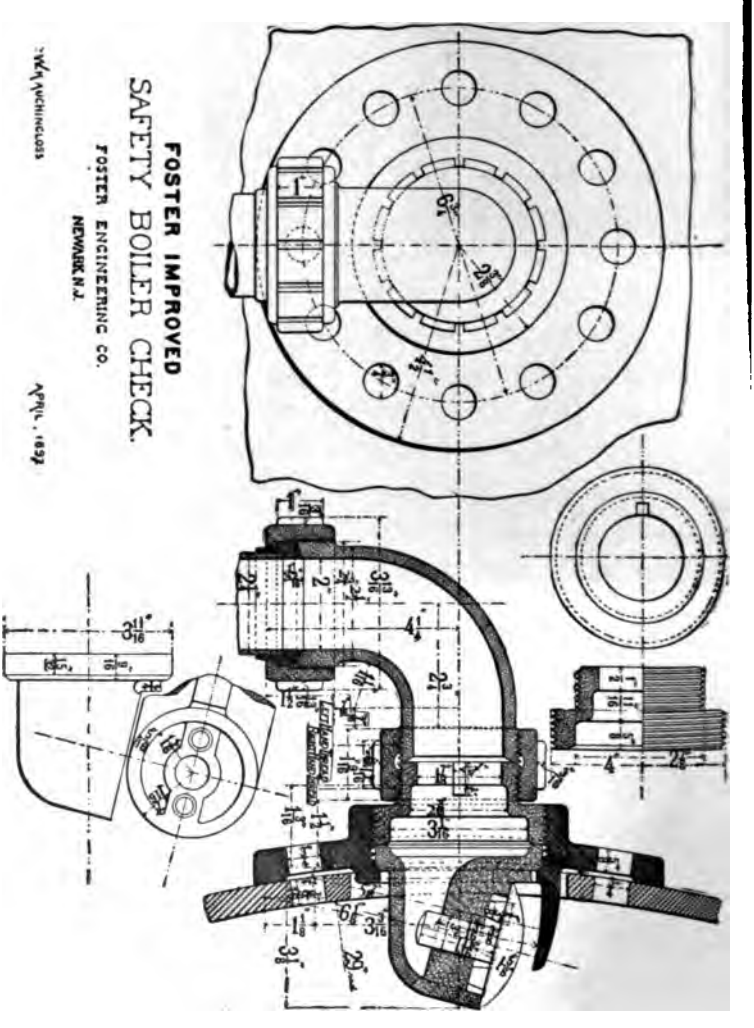
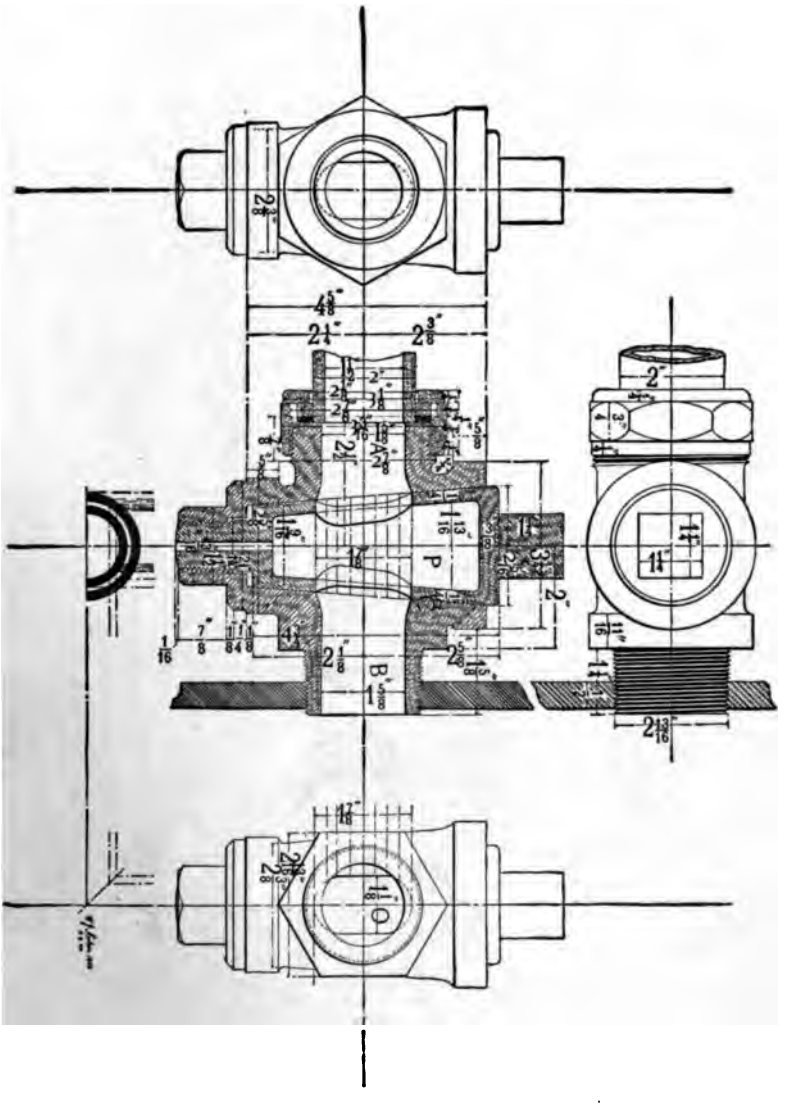

FOSTER IMPROVED
SAFETY BOILER CHECK.
FOSTER ENGINEERING CO.
NEWARK, N.J.
APRIL, 1897

FIG. 217.


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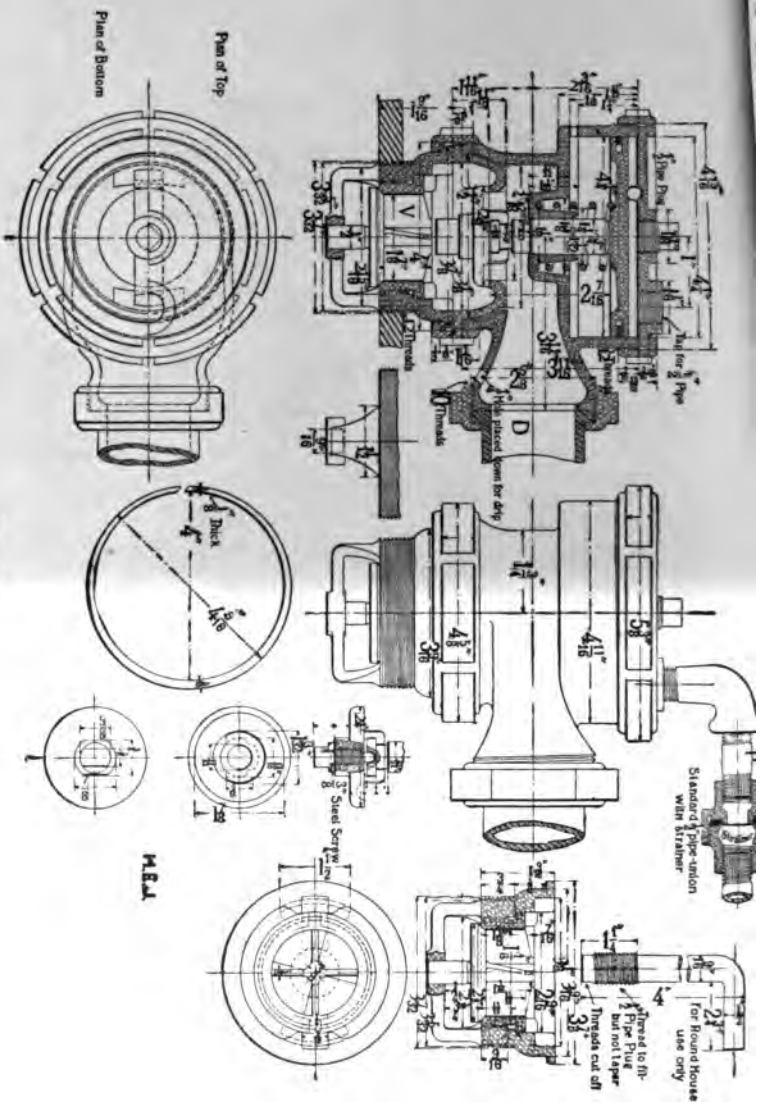

Plan of Top
Plan of Bottom
Standard pipe-union with Strainer
for Round House use only
Threads cut off
Hole placed here for drip
Threads
Duck
Steel Screw

FIG. 219.

erated by means of compressed air. The wing-valve *V* is held on its seat by the steam-pressure in the boiler. When compressed air is introduced into the cylinder *C* through the pipe *D* the piston is pushed against the valve, opening it and allow-

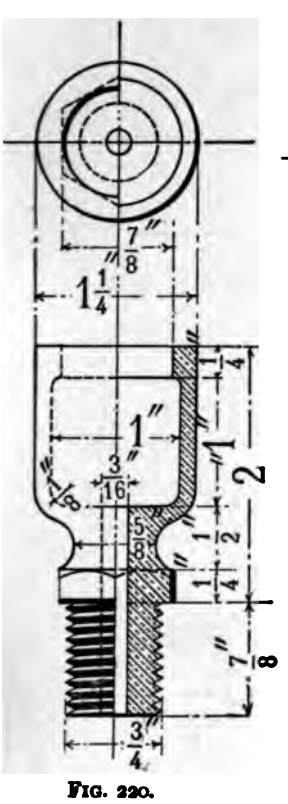


FIG. 220.

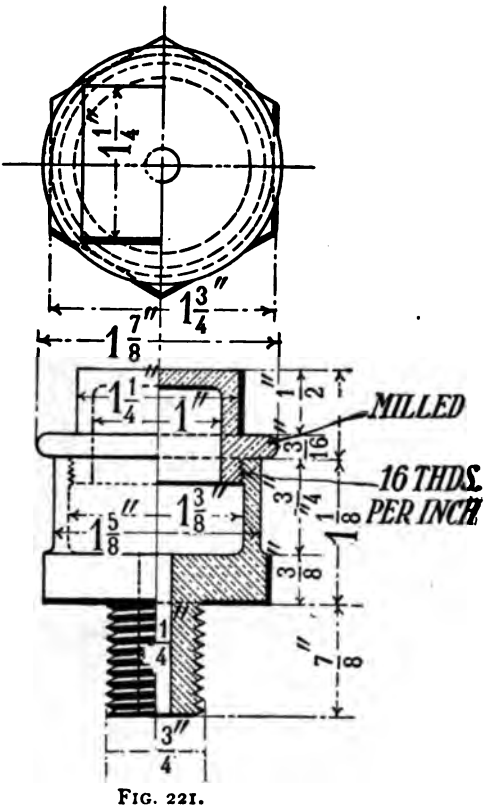


FIG. 221.

ng the contents of the boiler to blow through the cock into he discharge-pipe *D*.

Exercise 108.—Make complete drawings as shown in Fig. 19. Scale, full size.

Oil-cups.—There are many forms of oil-cups. Figs. 220

to 225 inclusive show the construction of some of the oil-cups used in the locomotives of the Lehigh Valley Railway.

Fig. 220 is one of the simplest forms of oil-cups. The material is brass, cast in one piece. When charged, the reser-

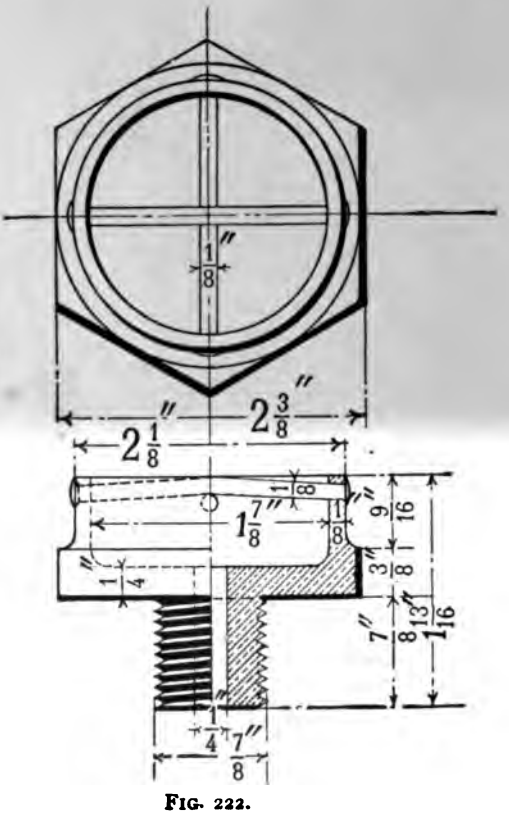


FIG. 222.

voir is filled with waste and oil. This cup is used on the link-hanger.

Fig. 221 shows another simple form of oil-cup, used to oil the rocker-box and cross-head.

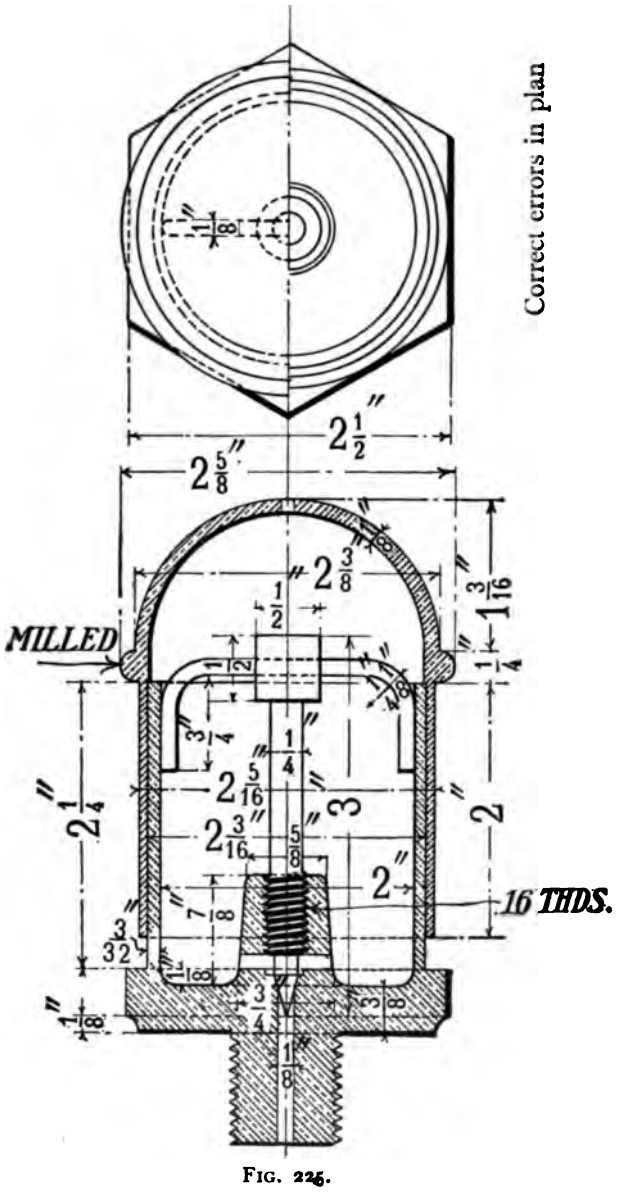


FIG. 225.

main rod on cross-head. It will be seen that in this case the flow of the oil is also mechanically controlled.

Fig. 225 is a form of cup used on the guides. The flow of the oil is in this case also regulated by the raising or the lowering of the spindle by hand.

Exercise 109.—Make drawings, as directed by the instructor, of one or more of the oil-cups illustrated in Figs. 220 to 225 when it is desired to fill unoccupied space on drawing-paper. *Scale, full size.*

CHAPTER X.

ENGINE DETAILS.

The Plain Slide-valve.—The construction of all slide-valves must be such as to satisfactorily meet the following requirements:

1. To admit steam to one end only of the cylinder at a time;

2. To allow the steam in the cylinder to escape from one end at least as soon as steam is admitted at the other end;

3. To prevent steam from entering the exhaust-port from the steam-chest.

During one revolution of the crank there are four principal points reached and passed by the valve in the course of its travel:

1. The *point of admission*, when steam begins to enter the cylinder. (See Fig. 236, Plate I.)

2. The *point of cut-off*, when steam is prevented from entering the cylinder. (See Fig. 233, Plate I.)

3. The *point of exhaust*, when steam is released from the cylinder. (Fig. 235, Plate I.)

4. The *point of compression*, when the exhaust is closed. (Fig. 234, Plate I.)

In Fig. 226 is given a longitudinal section of a plane slide-valve, and also of the valve-seat S of the cylinder. The valve is shown in its central position, X and Y are the steam-ports and Z the exhaust-port. The valve-face is the under side of the valve with a length equal to $F = 7\frac{1}{4}'' + 2L$.

Outside Lap, or simply *lap* is the darkened portion L of

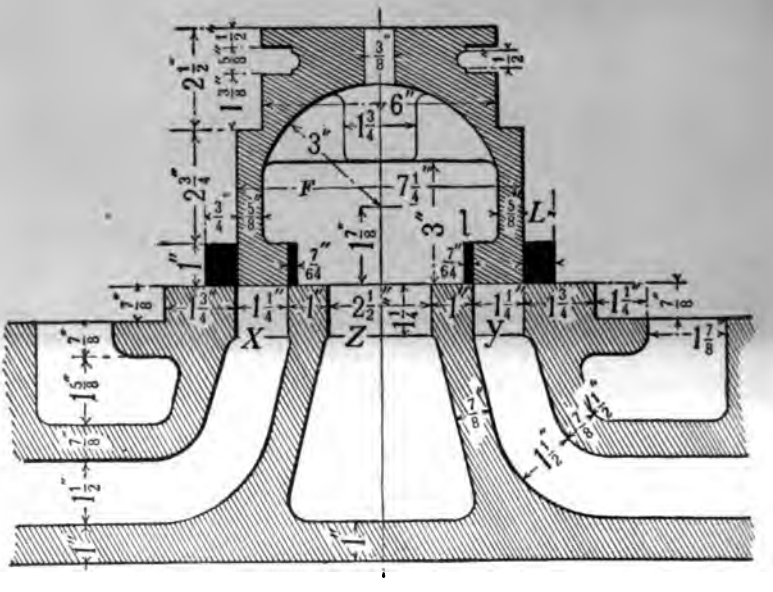


FIG. 226.

the valve which overlaps the steam-port when the valve is in its central position. Lap has no effect on compression or exhaust, but it hastens the cut-off, prolongs expansion, and shortens the time the port is open.

Inside Lap, the smaller darkened portion l which overlaps the bridge between the steam- and exhaust-ports, prolongs expansion, hastens and increases compression, retards

the exhaust, but does not affect the admission or point of cut-off.

The Travel of the valve is equal to twice the total distance it moves from its central position in either direction; or if the arms of the rocker are of equal lengths, then the travel of the valve is equal to twice the eccentricity of the eccentric. (See "Eccentric and Straps.") It is also equal to twice the sum of the width of the steam-port and lap plus the over-travel if any.

The Lead Angle is the angle made by the centre-line of the crank with the centre-line of motion of the engine when the crank is at the point of admission. (See Fig. 236, Plate I.)

The Lead is the amount which the valve has opened the steam-port at the beginning of the stroke. (See Fig. 231, Plate I.) To obtain smooth running, increased speed should have increased lead, and when the lead is increased every operation of the valve is quickened.

The Angle of Advance of the eccentric is the number of degrees which the centre-line of the eccentric is over 90° ahead of the centre-line of the crank without a rocker, and with a rocker it is the number of degrees short of 90° behind the crank. The first case is illustrated in the diagram in Plate II, as follows: Let *AO* be the centre of the crank, *CO* a line 90° ahead of it, and *OE* the centre-line of the eccentric. Then *COE* is the angle over 90° ahead of the crank, and is therefore the angle of advance. For the case with a rocker let *AO* be the centre of the crank as before, and *OD* a line 90° behind it, and *OF* the centre-line of the eccentric. Then the angle *DOF* is the angle short of 90° behind the crank, and is therefore the angle of advance.

Inside Clearance is the opposite of inside lap, instead of the valve overlapping the bridge when on the centre; as shown at *l* in Fig. 226, it shows a clearance between the inside edge of the valve and the bridge. Inside clearance hastens exhaust, delays compression, but has no effect on the cut-off or admission.

Overtravel is the distance the steam edge of the valve travels after fully opening the port, as shown in Fig. 232, Plate I. It increases the sharpness of the cut-off, retards compression, and gives a later release.

Cylinder Clearance is all that space between the faces of the piston and the valve when the piston is at the beginning of the stroke.

Piston Clearance is the distance between the piston and the cylinder-head. This clearance is to prevent the piston from striking either cylinder-head when the brasses on the connecting-rod wear and cause lost motion.

Point of Cut-off is the point on the crank-circle which the centre of the crank reaches when the valve cuts off the live steam from the cylinder, and for the remainder of the stroke utilizes the expansive power of the steam. (See Fig. 233, Plate I.)

Compression of the steam follows the closing of the exhaust before the piston has completed its stroke. This is done to obtain a yielding cushion for the reciprocating parts to come to a full stop without shock before beginning the return stroke.

Expansion begins at the point of cut-off and continues to the point of exhaust. (See Figs. 233 to 235 in Plate I.)

During this period the valve travels a distance equal to the outside lap plus the inside lap.

The Allen-Richardson Balance-valve.—This is one of the most popular combination slide-valves and is used on locomotives, stationary and large marine engines. Fig. 227 clearly shows the different parts used in the construction of this valve. The balance is effected by means of four rect-

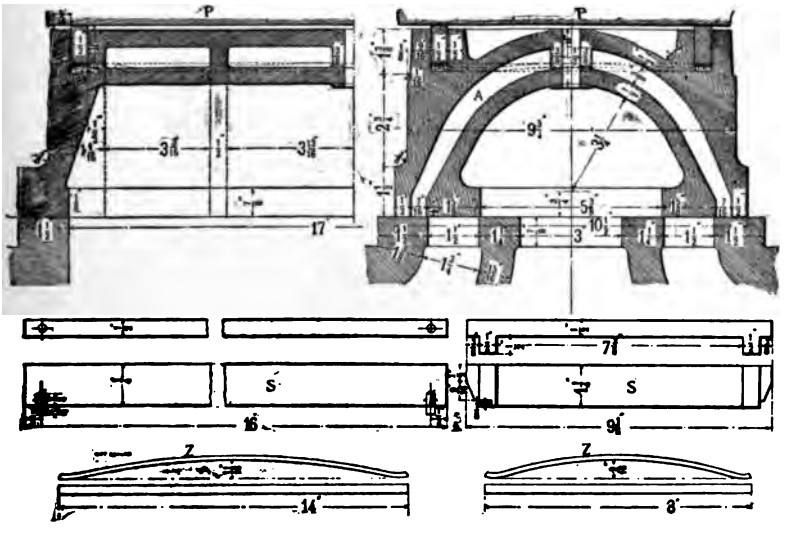


FIG. 227.

angular packing-strips *S* fitted into grooves on the top of the valve. Semi-elliptic springs *Z* are used to hold the packing-strips against the pressure-plate *P* when there is no steam in the chest, but when steam is admitted to the chest it forces the strips against the pressure-plate and sides of the grooves, forming a steam-tight joint and preventing the steam from acting on that part of the top of the valve enclosed by the four packing-strips.

Exercise 110.—Make drawings as shown in Fig. 227, and also a half plan of the top. Scale 8″ = 1 foot.

The Allen feature of this valve is the supplementary port shown at *A* just above the exhaust-arch. By means of this additional port steam is admitted to the same steam-port in the cylinder from both sides of the valve at the same time, thereby increasing the steam-supply with short cut-offs. The advantages of this valve over the plain slide-valve and the objections to it are the proceedings of the following societies: A. vol. 20, May 1899; The Western Railway Clul 897; Am. Railway M. M. Association, 1896; and ocomotive up to Date" by Chas. McShane.

*** The American de-valve.**—The American Balance is applied to any slide-valve. It consists of a steam-tight joint being to between the valve and the under side of the steam-chest cover, thus excluding live-steam pressure from a given area. (See Fig. 228.) This joint is formed by a bevelled snap-ring which, when in place, is slightly expanded over a cone. The cone or cones are either cast with the valves or bolted to it, as circumstances require.

The mechanical construction of the balance is: First, the cone or two cones, where necessity requires, are either bolted to or cast with the valve. The snap-rings, which are bevelled on their inner side to a corresponding degree with that of the cone, are bored smaller in diameter than their required working diameter so that, by their being forced down on the cone by the placing of the steam-chest cover in position, the rings

* The above description was furnished by Mr. J. T. Wilson, General Manager of The American Balance Slide-valve Co.

themselves are under tension and are thus supported by their own elasticity when not under steam. The steam when admitted to the steam-chest exerts a pressure on the entire circumference of the ring, which has a tendency to close it or decrease its diameter, and owing to its bevelled face and the taper of the cone the steam also acts to lift it. By careful



FIG. 228.

consideration of the operation of this ring, now being held by the steam-pressure tightly against the face of the cone, it will at once be seen that all lateral wear is avoided, and the ring moves as a part of the cone or valve itself. It will also be noted that the ring is absolutely compelled to assume its working position by the pressure on its circumference. When steam is shut off from the engine and the engine allowed to drift, as in locomotives, the valve is free to leave its seat until

the cone comes in contact with the cover. This affords perfect and ample relief of the air which the piston is forcing from one end of the cylinder, and also a direct communication with the other end of the cylinder, in which a vacuum is being formed. The cylinders are therefore perfectly relieved by allowing the valve to lift $\frac{1}{8}''$ off its seat.

The bevelled feature in the ring renders the ring self-supporting when not under steam, and supported by the steam-pressure when under steam, automatic adjustment for the wear, positive action under all conditions, and self-maintaining the steam-joint. It renders it possible also to duplicate the rings of respective sizes in repairs.



FIG. 229.

Owing to the absence of lateral wear on the cones new rings can be duplicated at any future time. The greatest area of balance can be secured by this design, because it is least affected by back or upward pressure. The valve in order to

leave its seat must first expand the taper ring against the chest-pressure acting on its circumference.

The features enumerated all depend upon the taper.

Fig. 228 is a double-cone balance-valve used on locomotives. The improved T ring, the invention of Mr. J. T. Wilson, is clearly shown in the figure.

Fig. 229 is a single-cone balance-valve for use on compound stationary engines. A double-cone valve of this kind is in use on the Japanese cruiser "Chtose," the rings of which

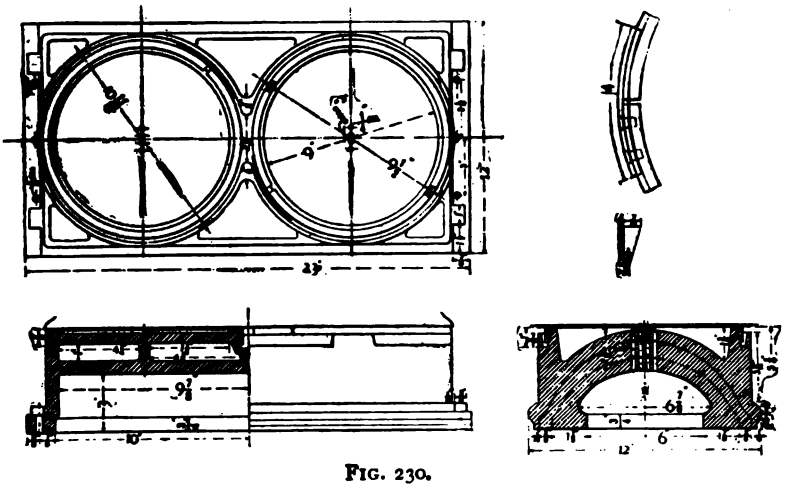


FIG. 230.

are three feet ten inches in diameter, while that in Fig. 229 is only twenty inches diameter.

Exercise III.—Make drawings of Fig. 230 as shown. Scale 6″ = 1 foot.

The Bilgram Diagram.—Among the many diagrams devised to determine quickly and accurately the position of the valve for any position of the crank, that due to Mr. Hugo Bilgram is one of the simplest and best.

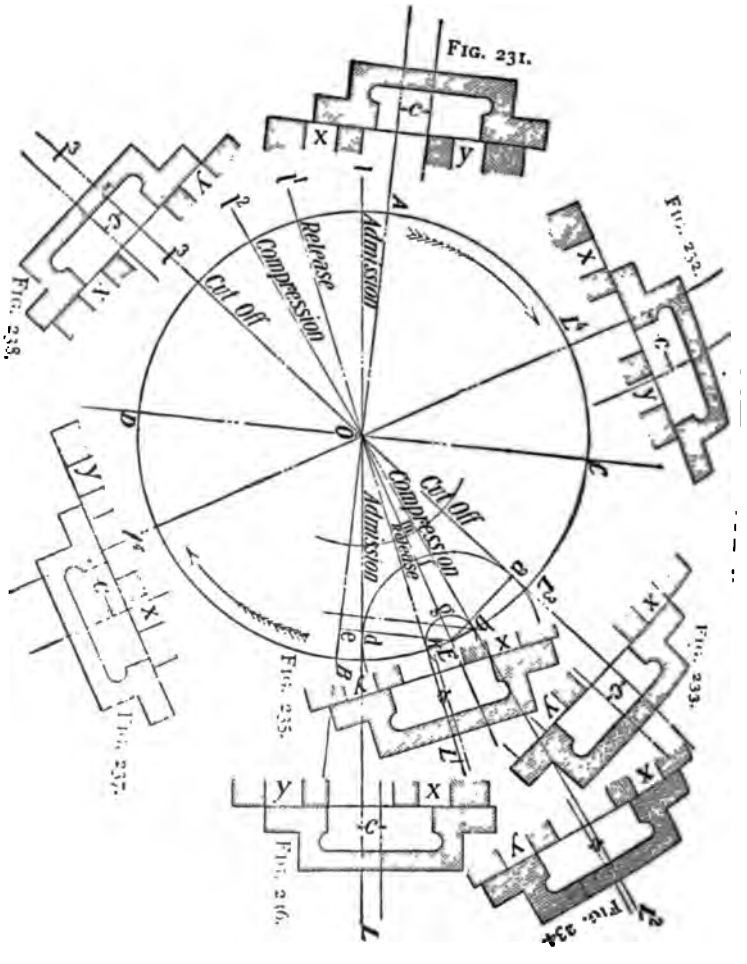

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FIG. 231.

In Plate I let *AB* represent the valve circle, equal in diameter to the travel of the valve, and *Ll* the centre-line of the crank rotating in the direction of the arrow. From *B* lay off the angle *EOB*, equal to the angle of advance. At *E* describe the arc *bgk* with a radius equal to the inside lap, and also the arc *afd* with a radius equal to the outside lap. Crank positions drawn tangent to these arcs at *a*, *b*, *k*, and *d* will give the points of cut-off, compression, release, and admission respectively, as indicated in the figure.

Let us follow the crank through one revolution, beginning with the dead-point *A*. In this position *de* is equal to the outside lead, and the valve has moved from its central position a distance *Ee* equal to the lap plus the lead. These relations are clearly shown in Fig. 231. *c* gives the distance which the valve has travelled from its central position, and at *X* the left-hand steam-port is shown open to steam an amount equal to the lead when the piston is at the beginning of its forward stroke, and the eccentric is connected directly to the valve, i.e., without a rocker.

When the crank reaches the position L^4 perpendicular to *OE* the valve will have travelled from its central position a distance equal to *EO*. This is the extreme position of its forward travel, as shown in Fig. 232. The maximum opening of the port *X* to steam is equal to *Of*, and the overtravel to *mf*, the actual width of the steam-port being = *Om*.

As the crank leaves L^4 the valve begins to return, and when the crank is at L^3 the distance of the valve from its central position is equal to the lap *ab*. Port *X* is now closed to steam, and cut-off is accomplished as shown in Fig. 233.

When the crank is at L^3 the right-hand steam-port is

closed to exhaust, and compression begins as shown at *Y*, Fig. 234.

When the crank reaches L^1 the valve is on the point of opening port *X*, Fig. 235, to release the steam which was under compression during the time the crank moved from L^2 to L^1. At crank position *L* we find, as shown in Fig. 236, that the valve is on the point of admitting steam to port *Y*, and at *B* the backward stroke of the piston begins, the valve having opened the port an amount equal to the lead *de*, equal to the opening shown at *X* in Fig. 231.

At crank position l^1 and valve position Fig. 237 the valve has attained its maximum travel in the opposite direction to that shown in Fig. 232. At l^2, Fig. 238, the valve cuts off steam from port *Y*, and at *A* the *new* forward stroke begins.

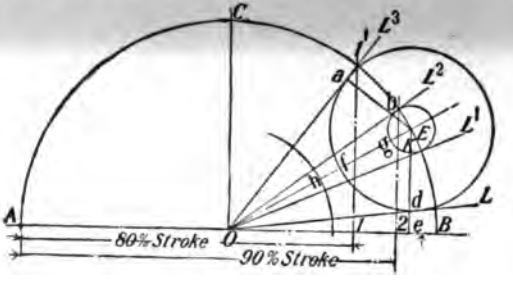


FIG. 239.

Exercise 112.—(Fig. 239.)

Given.		Required
Travel	= 5″.	Outside lap.
Angle of advance...	= 30°.	Inside lap.
Cut-off.....	= 80% of stroke.	Outside lead.
Compression.....	= 90% of stroke.	Inside lead.
Width of steam-port	= $1\frac{1}{4}$″.	Maximum port opening.
		Overtravel.

Draw *AB* and *CO* at right angles to one another. Describe the valve-circle arc *ACB* with a radius equal to half the travel or eccentricity of the eccentric = $2\frac{1}{8}''$ to the scale of twice full size. From *B* lay off the angle *EOB* equal to the angle of advance = 30°. Let *AB* represent the stroke, and from *A* lay off *Al* = 80% of a stroke of 24″, and erect a perpendicular to cut the valve circle in l^1. Draw OL^1 through l^1; this is the crank position at the point of cut-off. Through *E* perpendicular to OL^1 draw *Ea*. With centre *E* and radius *Ea* describe the lap circle *afd*. From *A* lay off *A*2 = 90% of the stroke, and erect a perpendicular to cut the valve circle at *b*. Through *b* draw OL^2, which is the crank position at the point of compression. With *E* as centre and *Eb* as radius describe the inside lap circle. Draw OL^1 tangent to *bgk* at point *k*.

At *O* with a radius = the port opening describe the arc *h*. Then *Ea* is the required lap, *Eb* the inside lap, *de* the lead, *ke* the inside lead, *Of* the maximum port-opening, and *hf* the overtravel.

Exercise 113.—(Fig. 240.)

Given.		Required.
Cut-off.....	= 80% of stroke.	Travel of the valve.
Lap.....	= 1″.	Angle of advance.
Lead.....	= $\frac{1}{4}''$.	

Draw to a scale equal to twice full size *AB* and *CO* at right angles. Draw OL^2, the position of crank at cut-off. Draw line 1 2 parallel to *AB* at a distance above it equal to the lead *ed*. Draw line 3 4 parallel to *AB* at a distance equal to the lap plus the lead above it. With a radius equal to the

given lap find by trial a centre on the line 3 4, and draw the lap circle *afd* tangent to OL^3, and line 1 2 at the points *a* and *d*.

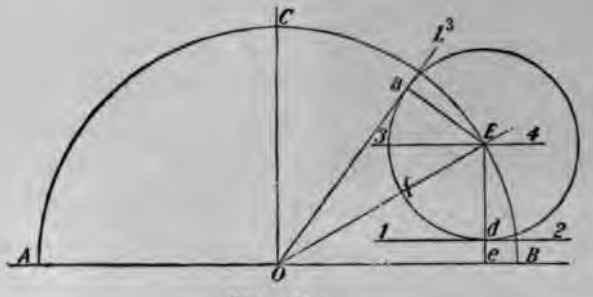


FIG. 240.

Then through *E* with centre *O* describe the valve circle *ACB*.

AB is the *travel* of the valve, and *EOB* the *angle of advance*.

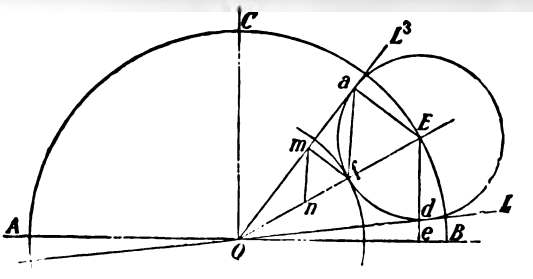


FIG. 241.

Exercise 114.—(Fig. 241.)

Given.		Required.
Cut-off	= 80% of the stroke.	Travel of the valve
Admission	= 90% of the stroke.	Lead.
Maximum port-opening	= *Of* (Fig. 240).	Angle of Advance
		Lap

Draw *AB* and *CO* at right angles. Draw *OL*, the position of the crank at the point of admission. Draw *OL*¹, the crank position at cut-off, and arc *f* with a radius equal to the given maximum port-opening. Bisect the angle *LOL*¹ with the line *OE*. The centre of the lap-circle will be on this line. Draw *fm* perpendicular to *OL*¹, and make *fn* = *fm*. Through *f* draw *fa* parallel to *nm*, and *aE* parallel to *mf*. *E* is the centre of *lap circle*. Through *E* describe the valve circle *ACB*, and draw the line *Ee* at right angles to *AB*.

Then *AB* is the *travel of the valve*, *Ea* the lap, *BOE* the *angle of advance*, and *de* the lead.

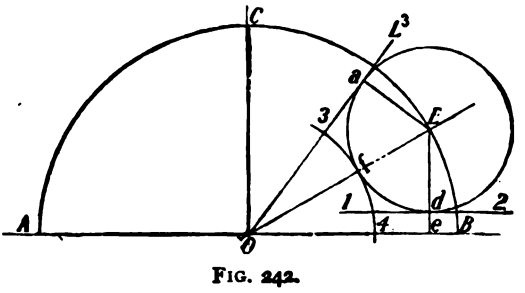


FIG. 242.

Exercise 115.—(Fig. 242.)

Given.		Required.
Cut-off.....	= 80% of the stroke.	Angle of advance.
Lead.....	= $\frac{1}{4}''$.	Lap.
Maximum port-opening	= *Of* (Fig. 240).	Travel of valve.

Draw *AB* and *CO* at right angles. Locate the crank position *OL*¹. Draw the lead-line 1 2 at a distance *de* from *AB* and parallel to it. With centre *O* describe arc 3 4 with a radius equal to the maximum port-opening. Find by trial the centre *E* of a circle that can be drawn tangent to *OL*¹, arc 3 4, and line 1 2. Through this centre draw *OE*.

Then *BOE* is the angle of advance, *Ea* the lap, and twice *OE* is equal to the travel of the valve.

The Zeuner Valve Diagram.—In Plate II let *AB* represent the stroke of the piston, the circle *ACBD* the path of the crank-pin, and *L* the centre-line of the crank.

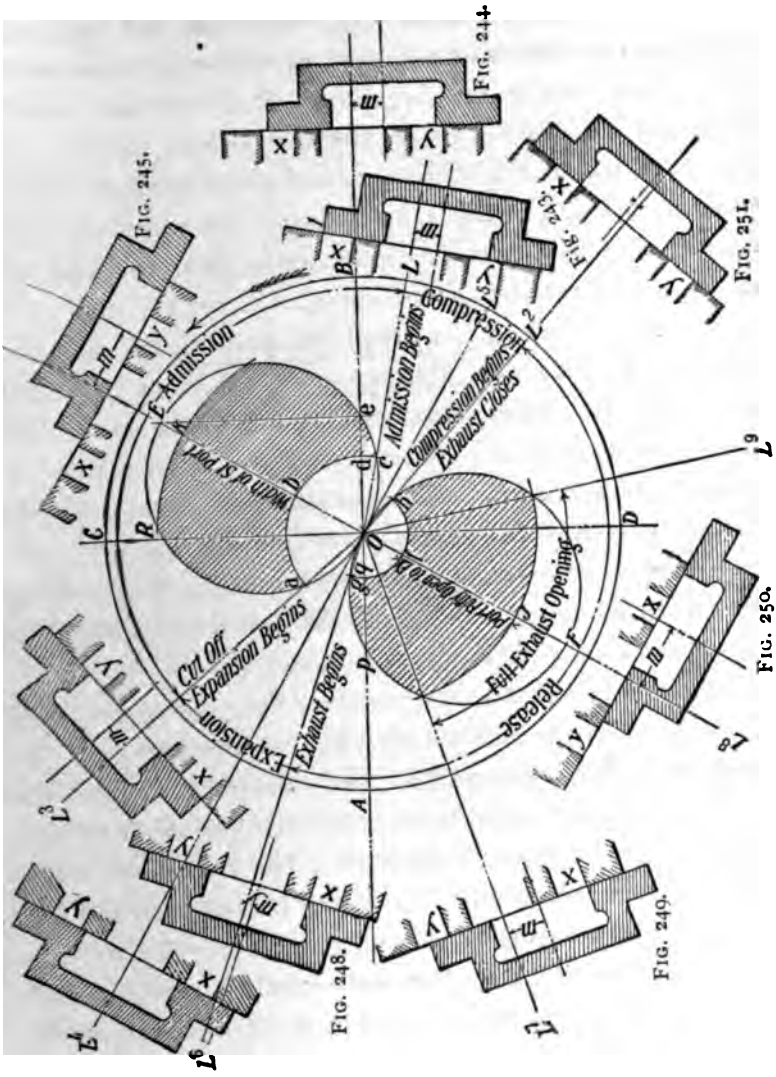
From *C* lay off *OE* equal to the angle of advance, and on *OE* as a diameter describe the valve circle equal to half the travel of the valve or eccentricity of the eccentric when no rocker is used. From centre *O* draw the arcs *abc* and *gfh* equal to the outside and inside laps respectively.

At the beginning of the forward stroke the true position of the crank would coincide with *AO*, and the centre-line of the eccentric with *OE*.

Now since the position of the point *E* is fixed for a given eccentricity and angle of advance, the point *E* will always be found on the circumference of the circle having *OE* as a diameter; and if the valve circle together with the crank be rotated around the centre *O* in a direction opposite to the arrow, its intersection with the line *OB* from *O* will be the distance which the valve has travelled from its central position after the crank has moved through any given angle.

But instead of rotating the crank and valve circle let them remain fixed and rotate the line *OB* as an imaginary crank in the direction of the arrow, and the same results will be obtained in a much simpler way.

Draw *OL*, the imaginary crank, through the point where the lap arc *abc* intersects the valve circle at the point *c*. The position of the valve will then be at the point of admission, because the valve will have travelled from its central position a distance equal to *Oc*, equal to the lap.


Admission
Admission Begins
Compression
Compression Begins
Exhaust Closes
Cut Off
Expansion Begins
Exhaust Begins
Exhaust Opening
Release
Pre-Exhaust-ing
Width of St. port
Port Face open to St.
Fig. 244.
Fig. 245.
Fig. 246.
Fig. 247.
Fig. 248.
Fig. 249.
Fig. 250.
Fig. 251.

This is clearly shown in Fig. 243. The valve is travelling in a direction opposite to the imaginary crank, and the steam edge *l* of the valve is on the point of admitting steam to the port *X* just before the beginning of the forward stroke. When the imaginary crank reaches the position *OB* the valve will have travelled a distance equal to *Oc* from its central position and, *OB* being a dead-centre, the valve will have opened the port *X* to steam an amount equal to the lead, and the port *Y* to exhaust an amount equal to *pq*, Fig. 244.

When the crank has reached the position *OE* the valve will then have attained the extreme position of its travel. The shaded part *bk* shows the full opening of the steam-port, and *ke* the amount of overtravel, and at the same time the port *Y* is fully open to exhaust, as shown by the shaded portion *fJ*, and *JF* is the exhaust overtravel (Fig. 245).

The valve now returns and at *R* begins to close port *X*, until when the crank arrives at L^3 the port is fully closed and cut-off takes place, as shown in Fig. 246.

When the crank is in the position L^4 at right angles to *OE* the valve is in its middle position, as shown in Fig. 247.

At the crank position L^5 the valve has travelled a distance *Og* from its central position, and the port *X* is about to open to exhaust, as shown in Fig. 248. The port continues to open, until at the position L^1 the port is fully open and continues so until the crank reaches the position L^6, when it begins to close, and is fully closed when the crank reaches L^2. Now compression begins and continues through the angle L^2OL. At *L* the valve has returned to the point of admission a little before the beginning of the new forward stroke.

At crank position L^6 it will be seen from Fig. 250 that the

port Y is fully open to steam, the port X fully open to exhaust, and that the valve has reached the extreme position of its travel for the backward stroke—just the opposite of the position shown in Fig. 245.

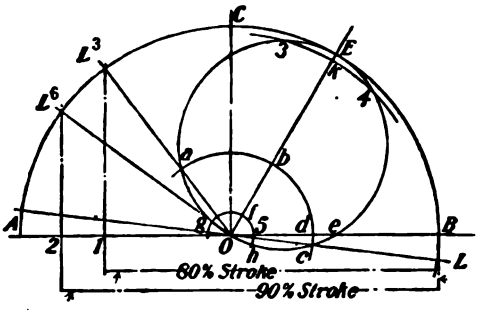


FIG. 252.

Exercise 116.—(Fig. 252). Assume the same conditions as in Ex. 112 for the Bilgram diagram. Draw AB and CO at right angles. Make AB to any convenient scale equal to the stroke of the piston, and let ACB represent the path of the crank-pin. From C lay off angle OE equal to the angle of advance = 30°, with a scale equal to twice full size. On Ok as a diameter, equal to half the travel of the valve, or $2\frac{1}{2}''$, describe the valve circle $Oakc$. From B lay off Bl equal to 80% of the stroke, and erect a perpendicular to cut the crank-pin arc in L^5. Draw OL^5, the position of the crank at cut-off. Through the point a, where OL^5 cuts the valve circle, with O as centre describe the arc abc. From B lay off $B2$ equal to 90% of the stroke, and erect a perpendicular to L^6. Draw OL^6, and through the point g, where OL^6 intersects the valve circle, with O as centre describe the arc gfh. From b lay off bk equal to the width of the steam-port, and with centre O and radius Ok describe the arc $3k4$.

Then *Oa* is the required lap, *Og* the inside lap, *de* the lead, 5*e* the inside or exhaust lead, *OE* the maximum port-opening, and *KE* the overtravel.

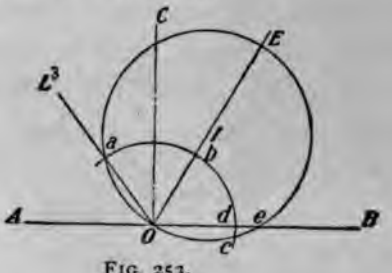


FIG. 253.

Exercise 117.—(Fig. 253.) Assume the same conditions as given in Ex. 113.

Draw *AB* and *CO* at right angles. Draw *OL*³, the crank position at cut-off. From *O* describe arc *abc* with a radius equal to the lap, scale as before. Lay off *de* equal to the lead. Bisect *Oa* and *Oc*, and the point *I* where the bisectors intersect will be the centre of the valve circle which may now be drawn through the points *aOe*.

Then *OE* is equal to half the travel of the valve, and *COE* is the angle of advance.

Exercise 118.—(Fig. 254.)

Given.	Required.
Point of cut-off... = 80% of stroke.	Travel of valve.
Point of admission = 90% of stroke.	Lap.
Lead..... = $\frac{1}{4}''$.	Angle of advance.

Draw *AB* and *CO* at right angles. Draw the crank positions *OL* and *OL*³. Bisect the angle *LOL*³ with the line *OE*. On *OE* assume any point as *g*, and draw *gf* perpendicular to *OB*, and *gh* perpendicular to *OL*. With center *O* and radius *Oh* describe the arc *he*.

Now the angles *gOB* and *gOL* are constant for a given admission and cut-off; therefore the lead will vary directly as the eccentricity.

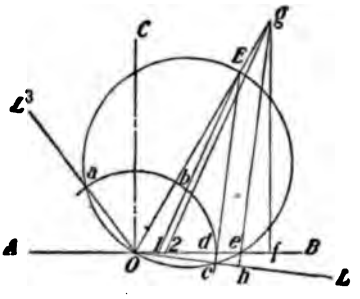


FIG. 254.

Let *Og* be an assumed eccentricity, then *ef* will be its corresponding lead, and the given lead is to the assumed lead *ef* as the required eccentricity is to the assumed eccentricity *Og*.

Lay off *Ol* equal to the given lead, and *O2* equal to *ef*. Draw *2g*, and *lE* parallel to *2g*.

With a radius equal to *Oe* minus the given lead and centre *O* describe the arc *abc*. On *OE* as a diameter describe the valve circle *Oaec*.

Then *COE* is the required angle of advance, *OE* the eccentricity or half the travel of the valve, and *Oa* the lap.

Exercise 119.—(Fig. 255.) Assume the conditions as in Ex. 115.

Draw *AB* and *CE* at right angles to each other. Draw *OL'*, the crank position at cut-off, *Oa*, the given lead, and *Ob*, the given maximum port-opening. At *b* erect the perpendicular *bg*. Through *a* draw *ad* at right angles to *OL'*. Bisect the angle *dce* with the line *Oc*, and produce it to intersect *fg* at *g*. Join

ag, and draw *cf* parallel to *Ob*. With *c* as centre and *cf* as radius describe arc *fk*, cutting *ag* in *k*. Join *ck*, and draw *a O* parallel to *kc*, cutting O_1g in the point O_1. Draw O_1e parallel to O_1A.

Then *Ock* is the required angle of advance, O_1a half the required travel of the valve, and O_1e the lap.

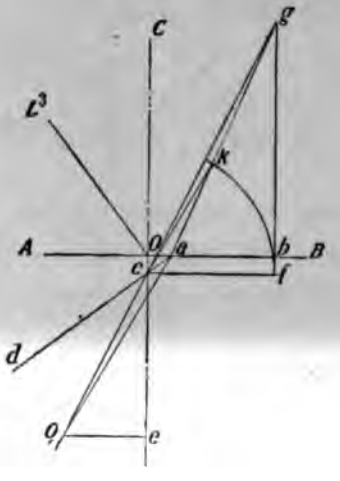
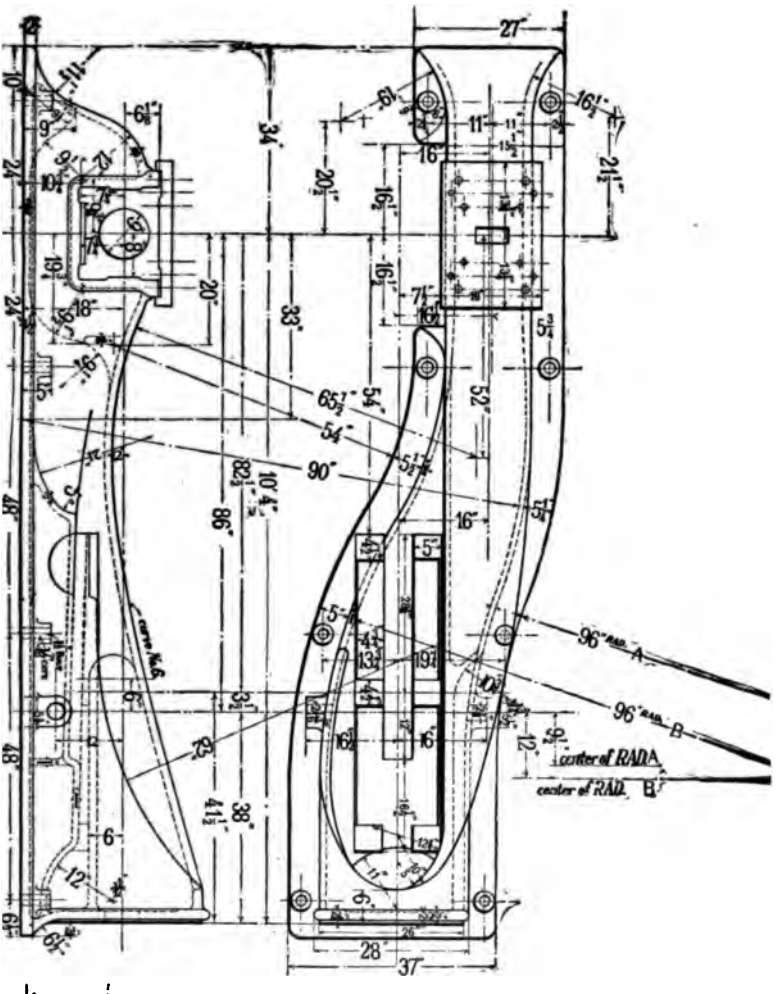


FIG. 255.

Engine Frame or Bed-plate.—Frames for horizontal engines are usually made of cast iron. This is the most suitable material owing to the complicated sections found in most frames, and also because it gives the necessary rigidity.

Figs. 256 to 258 show an engine frame of the "Tangye" type. It is that used by the Buckeye Engine Co. of Salem, Mass.

Exercise 120.—Make drawings as shown in Figs. 256 to 258. Scale $1\frac{1}{2}'' = 1$ foot.


27
34
20½
16½
21½
16½
20
33
65½
54
54
90
86
16
96 rad. A
96 rad. B
center of RADIA
center of RAD. B
28
37

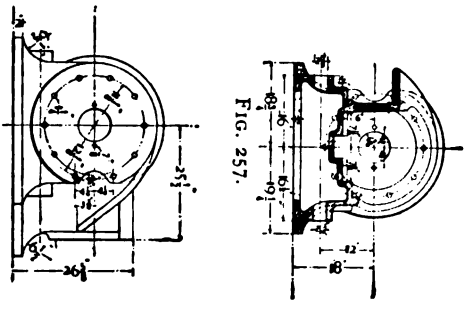


FIG. 257.

Cylinder.—Steam-engine cylinders are almost always made of a tough, close-grained cast iron as hard as can be safely worked.

Diameter of Cylinder, D.

Let P = the mean effective pressure of steam in pounds per square inch = M. E. P.;

L = length of stroke in feet;

A = area of piston in square inches;

N = number of strokes per minute.

Then $\frac{PLAN}{33000}$ = I.H.P. or indicated horse-power, and A is therefore $= \frac{\text{I.H.P.} \times 33000}{PLN}$; so $D = \frac{A}{.7854}$.

The mean effective pressure P may be found from the following formula:

Let p = the absolute initial pressure of steam, i.e., the gauge-pressure + 15 lbs.; − 7% for loss between boiler and cylinder.

r = ratio of expansion = length of stroke in inches ÷ distance travelled by piston in inches before steam is cut off.

Then $P = \frac{1 + \text{hyp. log. } r}{r}$ − back-pressure.

Thickness of Cylinder, t.

Let P_1 = boiler-pressure of steam per square inch in pounds;

D = diameter of cylinder in inches.

"Whitham" recommends the following formula for horizontal or vertical cylinders of large or small diameter where

rovision is made for reboring and sufficient strength and rigidity are secured:

$$t = 0.03\sqrt{P_1 D}.$$

Length of Cylinder.—The length of cylinder between eads = Stroke + thickness of piston + the sum of the piston learance at both ends.

Cylinder Head.—The cylinder head or cover next to the rank is sometimes cast on the cylinder.

The thickness of the cylinder head recommended by Prof. eaton is

$$\frac{P_1 D + 500}{2000}.$$

The thickness at the flange where the head is bolted to the ylinder should be $\frac{1}{8}$ greater than this.

Cylinder-head Bolts. — The diameter of cylinder-head uds in locomotives is usually $\frac{7}{8}''$, and their pitch about 4 mes their diameter.

For stationary and marine practice "Ripper" gives

$$n\frac{\pi}{4}d^2 f = \frac{\pi}{4}D^2 p,$$

here n = number of studs;

d = diameter of studs;

D = diameter of cylinder;

p = maximum pressure of steam;

f = 4000 to 5000.

Steam-ports.—The steam-ports which conduct the steam o.n the valve-chest to the cylinder should be as short and irect as possible, but large enough to prevent wire-drawing nd of easy curvature.

The length of ports in locomotives is usually 1″ less than the diameter of the cylinder.

In other types the length is generally made $0.8D$.

The area of the steam-port is given by many authorities as follows:

$$a = \frac{AV}{v},$$

where A = area of the piston in square inches;

v = velocity of steam through the port = 6000 feet per minute;

V = velocity of piston in feet per minute (from 1000 to 1200);

a = area of steam-port.

Figs. 259 to 262 show the working drawings of a horizontal steam-engine cylinder made by the Buckeye Engine Co. Fig. 259 shows a longitudinal section through the centre of the cylinder, Fig. 260 a cross-section through the exhaust-passage, Fig. 261 a back-end view showing the opening for a valve-rod, and Fig. 262 a plan.

Figs. 263 and 264 are the heads and covers suitable for this cylinder. When in position on the engine-frame the end of the cylinder, shown in Fig. 261, is bolted to the end of the frame shown in Fig. 261.

Exercise 121.—Make drawings of steam-cylinder as shown in Figs. 259 to 262. (*Scale* $1\frac{1}{2}'' = 1$ *foot.*)

Exercise 122.—Make the design of a cylinder similar to that shown in Figs. 259 to 262 to develop 100 I.H.P. Stroke 30″; steam-pressure 90 lbs. per square inch; cut-off at 50% of the stroke. Number of strokes per minute 220.

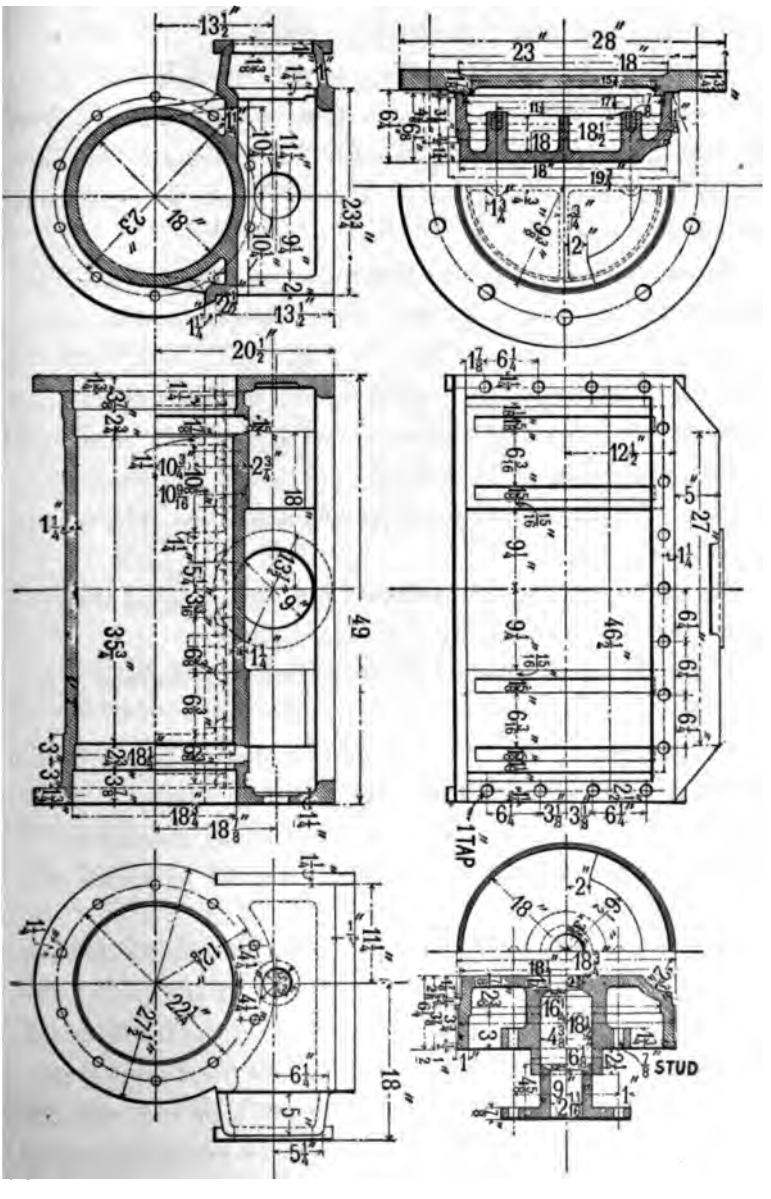


FIG. 263.

FIG. 262.

FIG. 264.

Exercise 123.—Make drawings of the cylinder heads shown in Figs. 263 and 264. (*Scale* $1\frac{1}{2}'' = 1$ *foot.*)

Pistons.—A piston is that part of an engine or pump which slides to and fro inside a hollow cylinder either driven by fluid pressure or acting against fluid pressure they are usually of circular section and are made of brass, wrought iron, cast iron, or steel.

A piston with valves which permit the fluid to pass from one side to the other is called a *bucket* and is used in pump cylinders.

A single-acting piston, guided by the stuffing-box instead of the cylinder, is called a *plunger* and is also used in pumps.

Steam-pistons.— A steam-piston should be designed so as to prevent the steam from passing from one side of the piston to the other.

The spring packing-rings should not press against the cylinder more than is necessary for steam-tightness.

A piston should be no heavier than is necessary for strength.

The weight of the piston should be distributed so as to prevent the excessive internal wear of the cylinder.

The piston must be firmly connected to the piston-rod.

Many different designs have been adopted to secure the above requirements.

Fig. 265 is a plain box piston, used by the Southwark Foundry & Machine Company. It is cast in one piece; the core being removed by three holes, shown in the front, which are afterwards plugged up. The two small holes are for eye-bolts which are used to remove the piston from the cylinder when necessary. The packing consists of two cast-iron spring

rings cut as shown in detail in the figure. The rod is forced into place by pressure and the ends riveted over.

Exercise 124.— Make drawings as shown in Fig. 265. (*Scale 6″ = 1 foot.*)

Fig. 266. This is another style of box pattern, used by the Ball Engine Company. The style of packing and the method of securing the piston-rod are plainly shown in the figure.

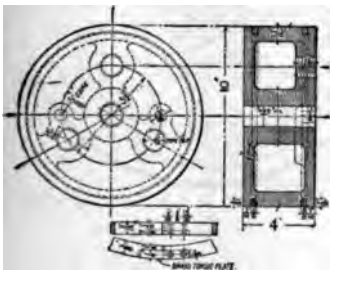


FIG. 265.

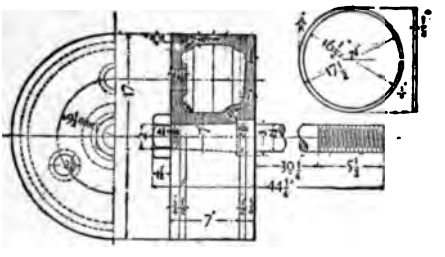


FIG. 266.

Exercise 125.—Make drawings as shown in Fig. 266, and in addition make a half end section through the centre of the piston. (*Scale 6″ = 1 foot.*)

Fig. 267 shows a common built-up piston used largely in locomotives. It consists of a *spider*, *S*, a T ring, a follower, *F*, and two cast-iron spring-rings. The rod is forced into place and held by nut over which the end of the rod is riveted.

Exercise 126.—Make drawings of a built-up piston like Fig. 267 for an engine whose cylinder is 18″ × 24″. Take dimensions from Table 36. (*Scale 6″ = 1 foot.*)

Fig. 268, a cast-iron box piston used in the cylinder of the Empire State Express locomotive. Its construction is plainly shown in the figure.

Exercise 127.—Make drawings of the piston shown in Fig. 268. (*Scale 6″ = 1 foot.*)

Fig. 269 is a cast-steel box pattern cast in two parts and

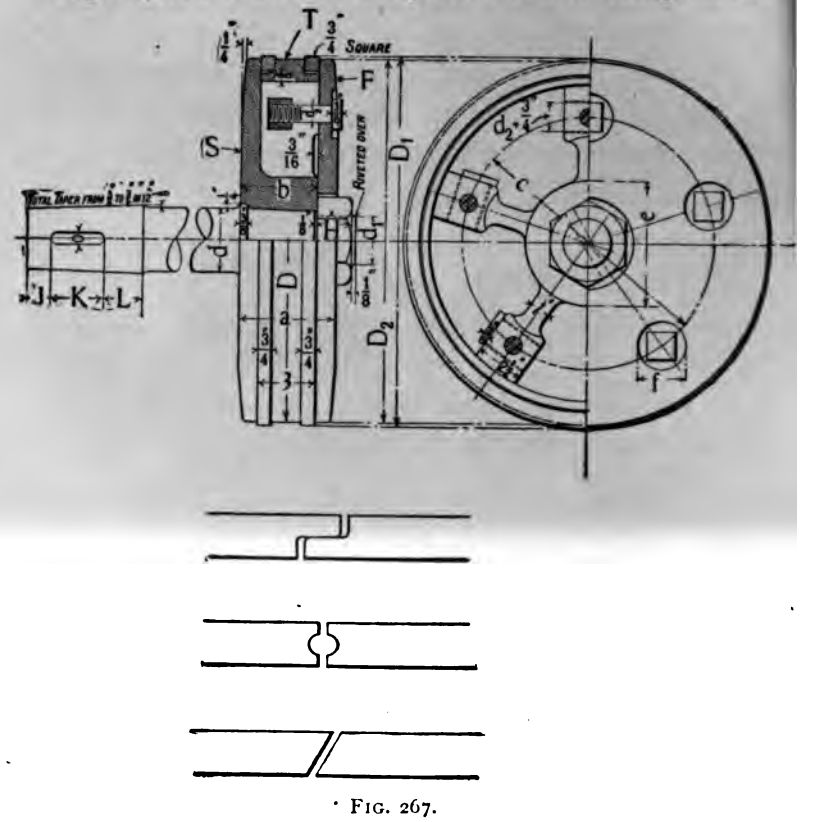


FIG. 267.

TABLE 36.

D	D_1	D_2	a	b	c	d	d_1	d_2	e	f	H	J	K	L	O
15	$15\frac{3}{16}$	$14\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{3}{4}$	10	$2\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$5\frac{7}{8}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$\frac{9}{16}$
16	$16\frac{3}{16}$	$15\frac{7}{8}$	5	4	11	$2\frac{3}{4}$	2	$\frac{7}{8}$	6	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{5}{8}$	$\frac{9}{16}$
18	$18\frac{3}{16}$	$17\frac{7}{8}$	5	4	13	$3\frac{1}{4}$	$2\frac{1}{4}$	$\frac{7}{8}$	$6\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$2\frac{5}{8}$	$1\frac{3}{4}$	$\frac{9}{16}$
19	$19\frac{1}{4}$	$18\frac{7}{8}$	5	4	14	$3\frac{1}{4}$	$2\frac{1}{2}$	$\frac{7}{8}$	$7\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$2\frac{3}{4}$	2	$\frac{9}{16}$
20	$20\frac{1}{4}$	$19\frac{7}{8}$	$5\frac{1}{4}$	$4\frac{1}{8}$	15	$3\frac{1}{2}$	$2\frac{3}{4}$	1	8	$2\frac{3}{4}$	2	$1\frac{1}{4}$	3	$2\frac{1}{4}$	$\frac{9}{16}$
22	$22\frac{3}{16}$	$21\frac{7}{8}$	$5\frac{1}{4}$	$4\frac{1}{8}$	17	$3\frac{3}{4}$	3	1	$8\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	3	$2\frac{1}{4}$	$\frac{1}{16}$

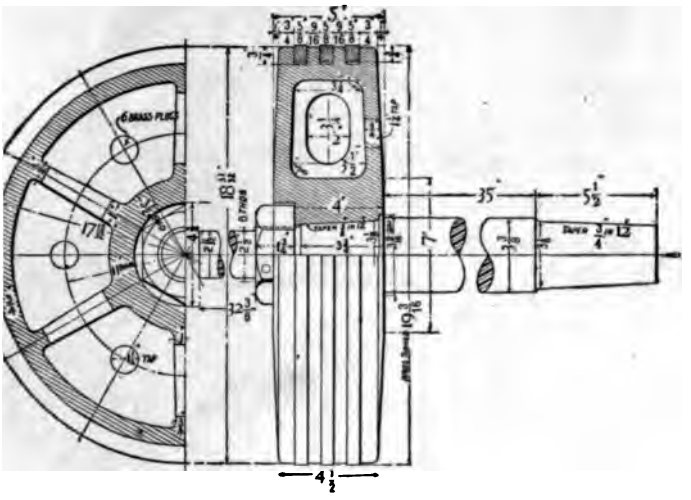


FIG. 268.

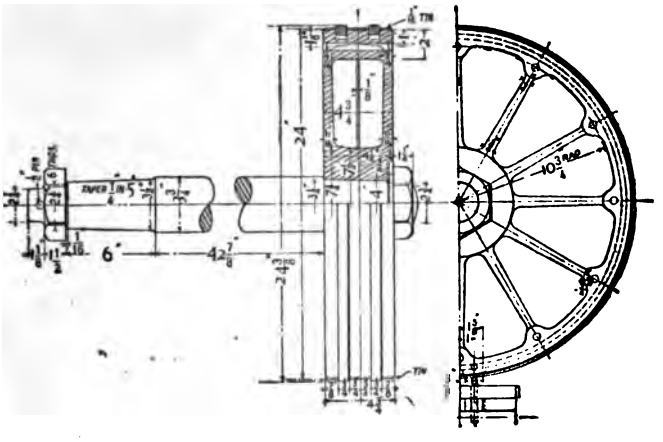


FIG. 269.

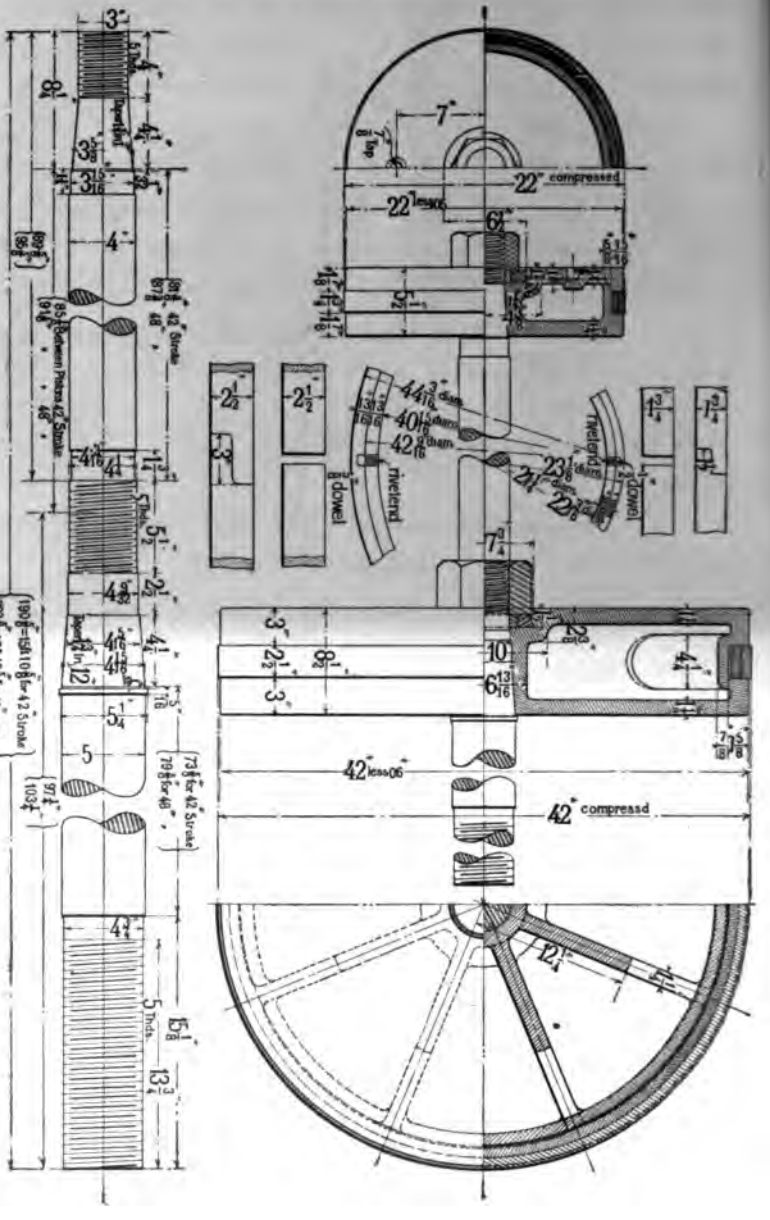

22" compressed
22" less 06
42" less 06
42" compressd
rivelend
dowel

FIG. 270.

held together by rivets. This piston is made by the Baldwin Locomotive Works for the "Vauclain" compound locomotive.

Exercise 128.—Make drawings as shown in Fig. 269. (*Scale 4″ = 1 foot.*)

Fig. 270 shows the cast-iron pistons used in tandem stationary engines built by McIntosh & Seymour.

The packing is composed of cast-iron spring-rings cut and kept in place by the method shown in detail in the figure. The arrangement for securing the rod is shown in detail in Fig. 68, page 103.

Exercise 129.—Make drawings as shown in Fig. 270. (*Scale 3″ = 1 foot.*)

Fig. 271 is a built-up piston for the Tangye stationary engines made by the Buckeye Engine Co. It consists of a spider, follower, and adjusting-screws. There are no springs; the screws act on an uncut junk-ring, so can only be used for centring, not for packing. The packing-rings are turned larger than the bore of the cylinder so as to pack by their own elasticity. They may or may not be turned eccentric, that is, thin where cut, and full thickness opposite the cut.

If made eccentric, it is for the reason that they will be more nearly round when sprung into the cylinder.

Exercise 130.—Make drawings as shown in Fig. 271. (*Scale 6″ = 1 foot.*)

Fig. 272 shows a water-piston suitable for cylinders under 9″ diameter.

The piston-rod is fitted to the head with a shoulder to drive the piston, and the rod is secured in place by a nut. The follower is also held by a nut and lock-nut. By this

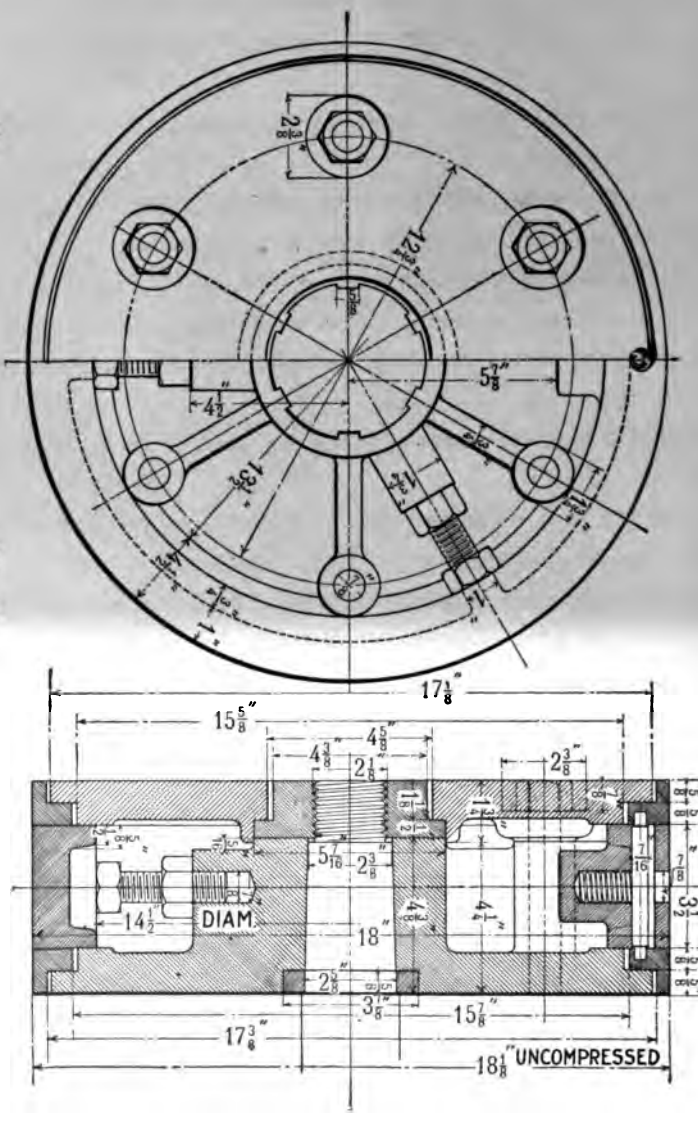


FIG. 271.

ans the follower and packing may be adjusted or renewed will. The packing is made of layers of cotton cloth and et rubber.

Exercise 131.—Make drawings of water-piston as shown Fig. 272. (*Scale full size.*)

Connecting-rods.—In steam and other engines the conting-rod connects the rotating crank with the reciprocat- cross-head.

There are many styles of connecting-rods, and various thods are employed for taking up the wear of the brasses. s. 273 to 276 show good examples of rods used in station-, locomotive, and marine engines of the most modern es.

Fig. 273 is the rod used by the Buckeye Engine Co. for ir "Tangye" type of engine. The crank end is *solid*, the sses are lined with babbitt, and adjustment for wear is had means of a tapered steel block and screws. The cross- d end is called a *strap* end. The strap is firmly bound to end of the rod with a cotter-key and gib, which also cons the adjustment for wear.

Fig. 274 has strap ends front and back. Keys are ined between the straps and the rod to prevent the shear of strap-bolts. The construction of this rod and the method ployed to take up the wear are plainly shown in the figure. e Erie City Iron Works use this rod on their stationary ines.

Exercise 132.—Make the drawings as shown in Fig. 273. *tle 6" = 1 foot.*)

Exercise 133.—Make the drawings as shown in Fig. 274,

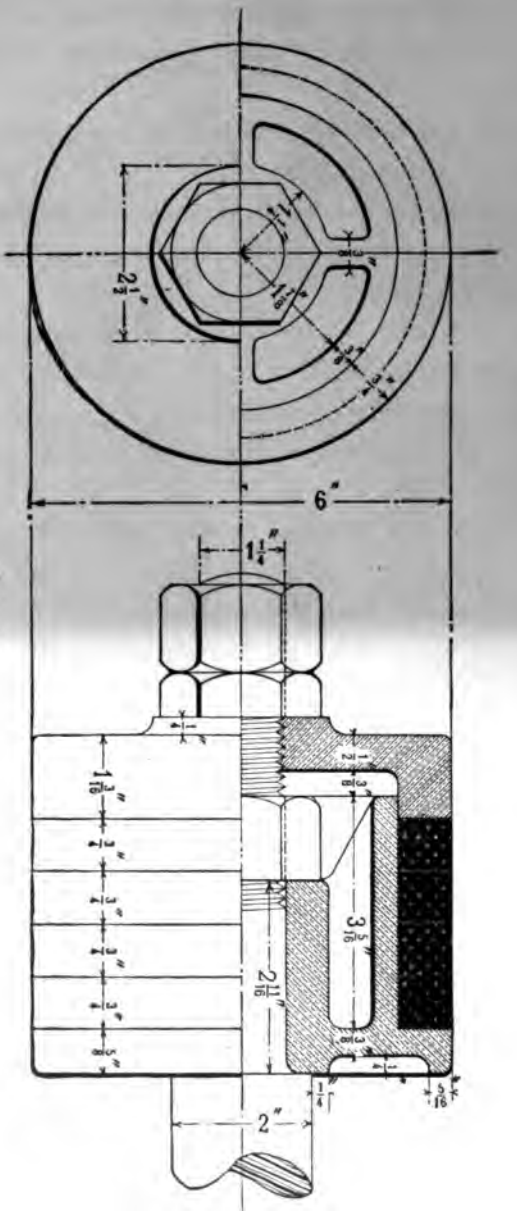


FIG. 272.

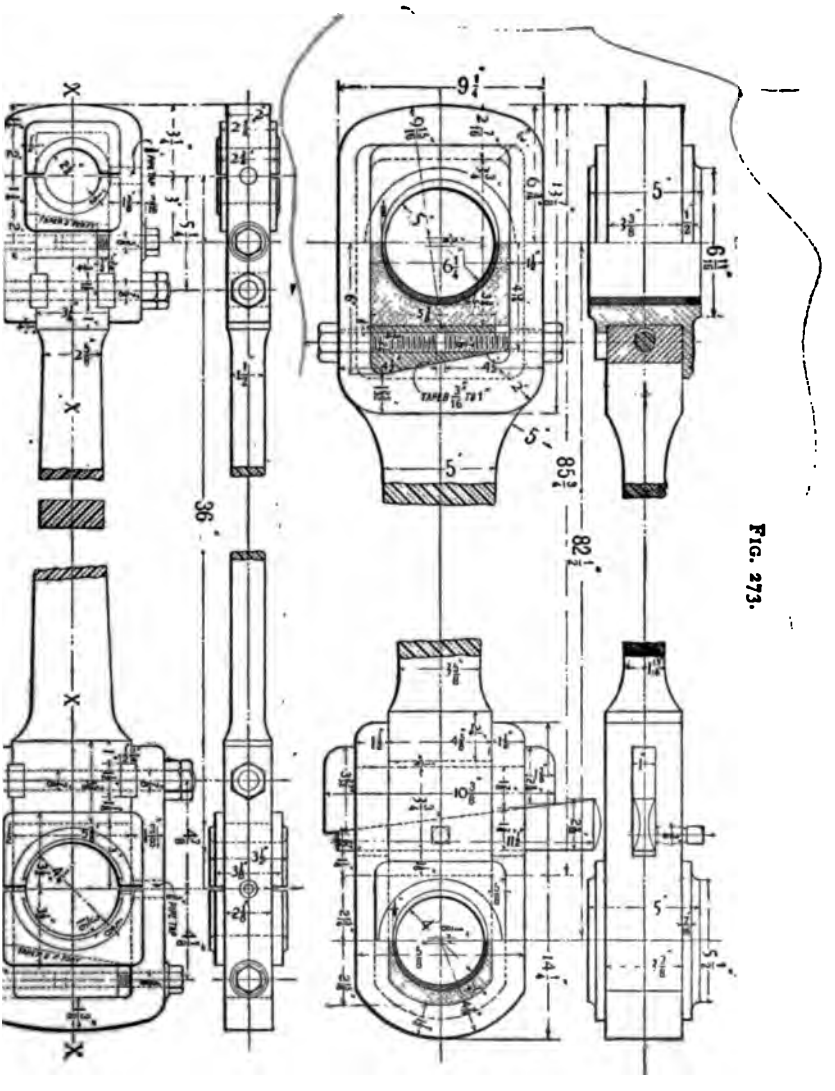
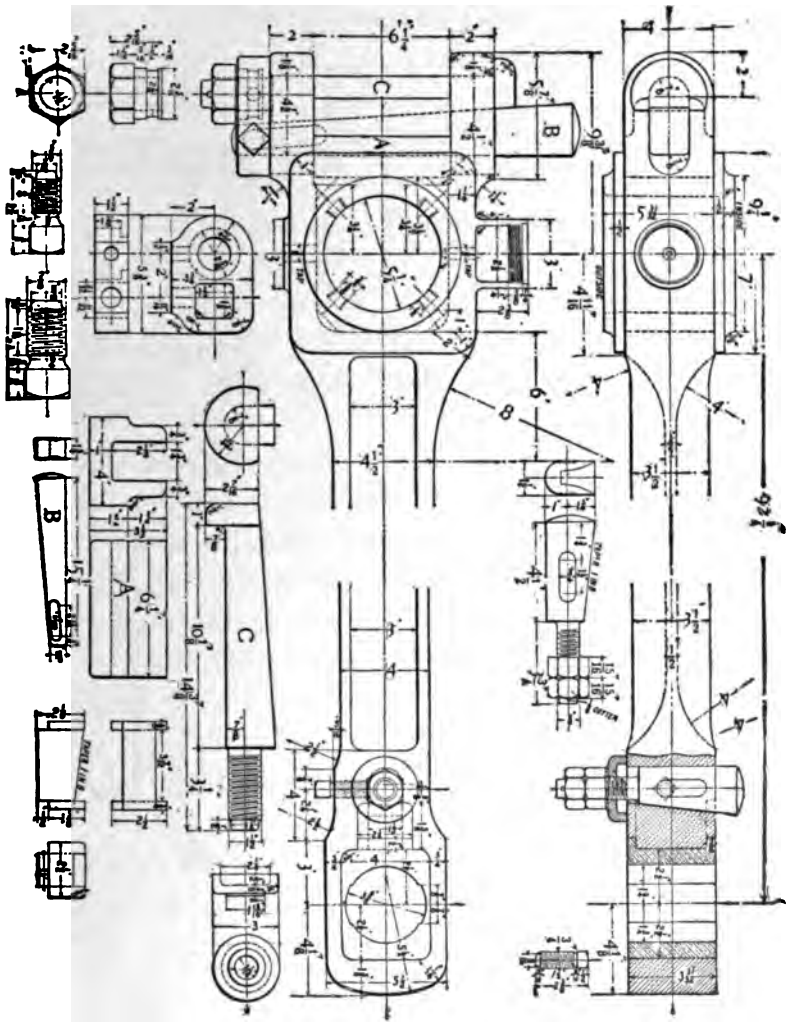


FIG. 273.

except that half of the plan shall be a section through *XX*. (*Scale* 6″ = 1 *foot*.)

Fig. 275 is the connecting-rod used by the Pennsylvania Railroad Company on their fast passenger-locomotives. The crank end of this rod is an improved design invented by Mr. A. S. Vogt, mechanical engineer of the company. He explains the improvements as follows:

As before, the back end of the rod is forked, but the method of closing the open end of the fork is entirely different, and the key for closing the main brasses has been moved from the forward side of the brass to the rear, which has another good effect, viz., as the brasses in both front end and back end of the rod wear and are closed up to meet that wear, the actual length of the rod changes but very little, for the reason that the keying of both ends is in the same direction, whereas in the old form of the rod the keying was in opposite directions, and as a matter of course the distance from centre of crank-pin to centre of crosshead pin increased gradually. The open end of the fork in this rod is closed, first, by a U-shaped block, the detail of which is marked *A* on Fig. 275; next, by the key which is marked *B*; and last of all by a combined key and bolt marked *C*; this bolt clamping the two members of the fork against the block *A* and forming an enclosed surface for the key to drive against. To prevent the slacking up of the nut *C*, a keeper-block is provided at the bottom of the lower member of the fork. This is made with a recess into which the nut fits and a set-screw for locking the nut. The same keeper-block extends forward to the key *B*, which is also blocked by a set-screw in the block. It is quite evident that there is much less chance of shearing



or offsetting of the bolt and the key in this than there was of the bolt in the former design; but even if it should take place, which is not very likely, the whole thing can readily be disconnected by, first, driving the key *B* out, unscrewing the nut on the bolt and moving the whole bolt slightly forward, when it can be lifted out at the top.

Exercise 134.—Make drawings as shown in Fig. 275. (*Scale 6″ = 1 foot.*)

This form of rod is called a marine connecting-rod because it is often used on marine engines, but it is also largely used on stationary engines and is occasionally seen on locomotives.

The crank-pin end or *stub* is usually forged solidly on the rod, and all but the sides is finished by turning in the lathe. The sides are then planed and the bolt-holes drilled. The hole to receive the brasses may now be bored unless the top and bottom of the brasses are to be thicker than the sides, in which event the hole will not be completed until after the cap or top end of the stub has been slotted off and bolted on again.

It will be seen that the bolts are turned down to a diameter equal to the diameter at the bottom of the threads. This does not weaken the bolt, but makes it more elastic.

The cross-head end of this rod is made forked to suit the cross-head, but it will be seen that each half of the forked end is constructed the same as in the large end.

A detail drawing of the bolt and its locking arrangement is given in Fig. 65, page 96.

Exercise 135.—Make drawings as shown in Fig. 276. (*Scale 2″ = 1 foot.*)

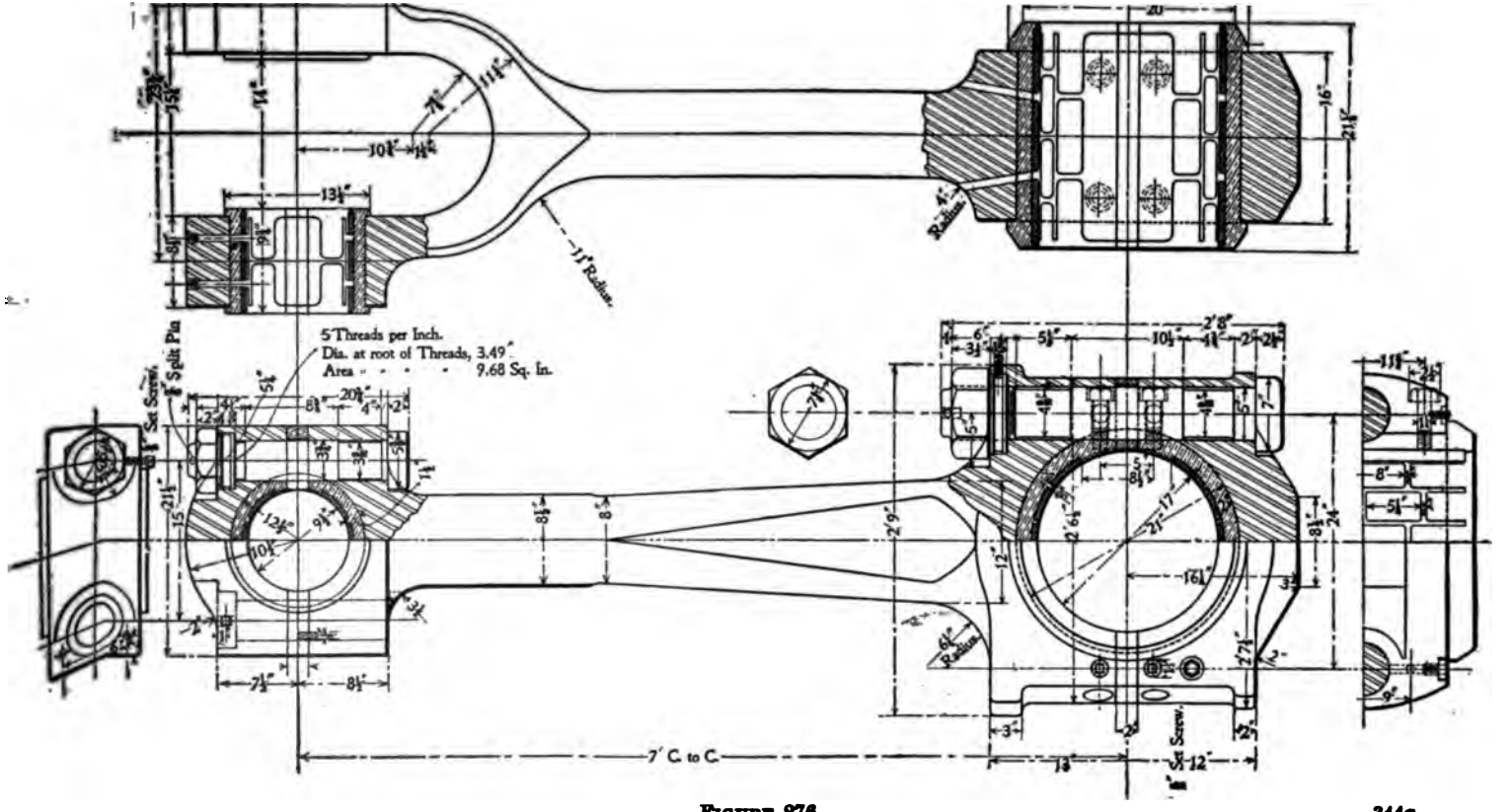

5 Threads per Inch.
Dia. at root of Threads, 3.49"
Area - - - - 9.68 Sq. In.
7' C. to C.

FIGURE 276.



Thrust of Connecting-rod.—Assuming that a connecting-rod is equal to a pillar rounded or jointed at both ends, let D = diameter of piston in inches;

L = length of stroke in inches;

l = length of connecting-rod in inches;

P = maximum steam-pressure per square inch;

T = thrust of connecting-rod.

When the crank-pin is on a dead-centre and the connecting-rod is in line with the piston-rod, then

$$T = \frac{\pi}{4} D^2 P = W,$$

the total load on the piston. But as the crank rotates the connecting rod becomes inclined to the centre line of motion, and T increases as the angle of the connecting-rod increases until a maximum is reached at half-stroke, provided the steam is not cut off before.

The value of T may be found for any position of the crank as follows:

Let AB, Fig. 277, be the connecting-rod, and BC the crank. The forces acting at A are W, the maximum pressure on the piston, and R, the reaction of the guide on the cross-head, and T, the thrust along the connecting-rod.

From the triangle of forces

$$\frac{T}{W} = \frac{AB}{AC}$$

and

$$T = W\frac{AB}{AC} = W\frac{AB}{\sqrt{AB^2 - AC^2}} = W\frac{l}{\sqrt{l^2 - \frac{L^2}{4}}} = W\frac{2l}{\sqrt{4l^2 - l^2}}.$$

Diameter of Connecting-rod, Circular Section.—Thurston gives

$$d = a\sqrt{Dl_1\sqrt{P}} + C = \text{diameter at middle,}$$

where $a = \begin{cases} 0.15 \text{ for fast engines,} \\ 0.08 \text{ for moderate speed;} \end{cases}$

$C = \begin{cases} \frac{1}{2}'' \text{ for fast engines,} \\ \frac{3}{4}'' \text{ for moderate speed;} \end{cases}$

$l_1 =$ length of connecting-rod in feet.

Seatons, Marks, and Whitham give

$$d = 0.02758\sqrt{Dl\sqrt{P}}.$$

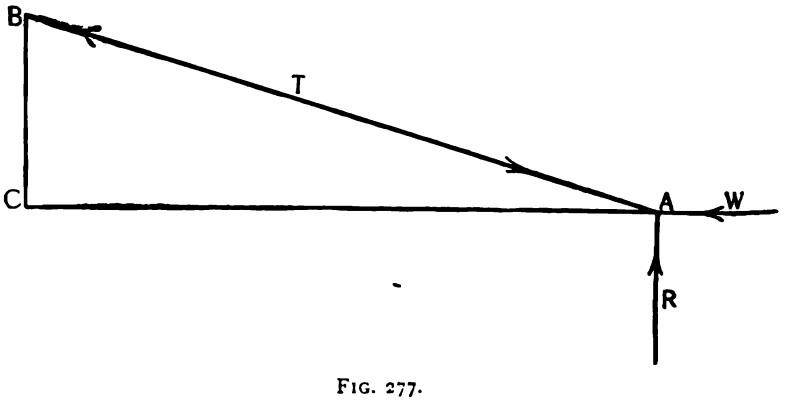


FIG. 277.

For the diameter at the crank-pin end Whitham gives 1.08 times the diameter at the cross head end. The rod is larger at the middle and tapers about $\frac{1}{8}''$ to the foot.

Sennett gives diameter at middle $= \frac{D}{55}\sqrt{P}$;

" " necks $= \frac{D}{60}\sqrt{P}$.

Locomotive Connecting-rods.—The sizes of rectangular rods of uniformly tapered section are in practice as follows:

Depth of Main Rod.—On engines with cylinders 14″ diameter or less the depth of the rod at the crank end is made $\frac{5}{8}$″ less than the depth of the stub; over 14″ diameter, $\frac{7}{8}$″ less.

Depth of main rod at cross-head end $= d_1$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
d_1	$2\frac{3}{4}$″	$2\frac{3}{4}$″	3″	3″	3″	3″	$3\frac{1}{4}$″

Thickness of main rod $= t_1$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
t_1	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{7}{8}$″	$1\frac{7}{8}$″	2″	2″	2″

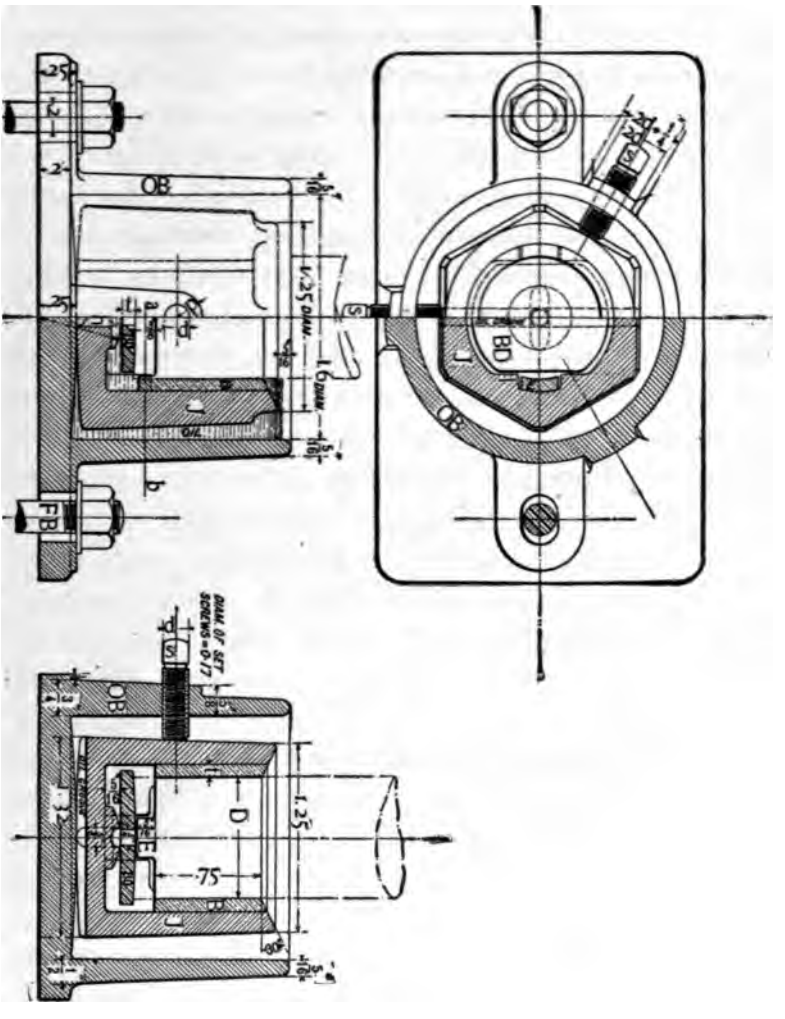
Depth of Side-rods.—The depth of the side-rod is made about $\frac{5}{8}$″ narrower than that of the stub end, and of uniform depth throughout.

Thickness of side-rods $= t_2$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
t_2	$1\frac{5}{8}$″	$1\frac{5}{8}$″	$1\frac{5}{8}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″

Pivot or Step Bearing.—In this form of bearing the pressure is applied in the direction of the axis, and the load is carried entirely upon the end of the shaft. In lubricating a bearing of this type the oil should always be introduced between the bearing surfaces from the under side and in the centre of the bearing, that the oil, under the influence of the

centrifugal force, will be distributed over the entire rubbing surface. The best results are obtained from running the bearing in a bath of oil, as shown in Fig. 278, where the oil-basin surrounds the journal-box, which can be kept submerged in oil, thus insuring constant and efficient lubrication. In Fig. 278 the oil passes from the oil-basin *OB* to the journal-box through the hole *h*, and on to the under side of the shaft through the hole in the disk *BD*. To insure a continuous flow of oil through the holes, grooves are made on the under side of the journal-box, and upon the upper side of the projection upon which the disk *BD* is carried. To prevent lateral motion the end of the shaft is turned to fit the journal-box *J* which is provided with a brass bush (*B*). The bush is secured against turning with the shaft by the projections *E*, which fit between lugs cast on the inner surface of the journal-box. The under side of the journal-box *J* is slightly spherical and rests upon a surface which is also slightly spherical. This allows the journal-box to be tipped over for a limited distance by means of the set-screws *S* until its axis coincides with that of the shaft. The down-pressure of the shaft is carried upon a steel disk *BD*, which also rests upon a slightly spherical projection cast on the upper side of the journal-box bottom, allowing the whole of the shaft end to remain in contact with the disk although the bush *B* has become sufficiently worn to allow the shaft to have side motion, thus making the bearing adjustable, and capable of maintaining a perfect bearing over its entire surface under all ordinary working conditions. The disk *BD* is longer than the shaft diameter, and is prevented from turning with the shaft by the flat sides coming in contact with the lugs *l*. The journal-box is hexagonal


OB
FB
BD
1·25 DIAM.
·16 DIAM.
·75
1·25
·32

in cross-section, and is tapered towards the top to allow it to tip the required distance without increasing the diameter of the oil-basin. It is held against turning with the shaft by the set-screws S pressing against the flat faces. This form of pivot or step bearing is suitable for journals from $1\frac{1}{2}''$ to $3\frac{1}{2}''$ in diameter.

As the velocity of the bearing surface varies from zero at the centre to a maximum at the circumference, and as the friction increases with the velocity, the wear will increase from the centre to the circumference. Thus it will be seen that the smaller the diameter D of the journal, within limits determined by the pressure per square inch on the rubbing surfaces, the more will the tendency to wear be reduced.

D will be found by the formula

$$\frac{\pi D^2}{4} = \frac{T}{P},$$

from which

$$D = \sqrt{\frac{T}{.7854P}}, \quad \ldots\ldots\ (1)$$

where P = intensity of pressure per square inch of projected area, which with this form of bearing running continuously may be taken at 300 lbs.;

T = total load on the rubbing surface, which is the weight of the shaft and its attachments.

Exercise 136.—Design a bearing of the form shown in Fig. 278, to carry a load of 1450 lbs. Show a HALF ELEVATION, a HALF SECTIONAL ELEVATION, a SECTIONAL END VIEW, the planes of section passing through the centre of the bearing, a HALF PLAN and a HALF SECTIONAL PLAN, the plane of sec-

tion passing through the bearing at the line *ab*. The unit of proportions $= D + \frac{1}{8}''$. The thickness t of the brass bush, and the bearing disk at the centre may be made $= .08D + \frac{1}{16}''$. The parts dimensioned in inches are constant for all sizes of journals. *Scale full size.*

Crank-shaft or Main Bearings.—The bearings carrying the crank-shaft of a vertical engine have the greatest pressure acting nearly vertically; consequently the greatest wear will be above and below the shaft, and adjustment is effected by a two-part bearing, parted on the horizontal centre line, as in Fig. 170. The crank-shaft bearings of horizontal engines should be designed for horizontal adjustment to take up the side wear caused by the pull and thrust transmitted along the connecting-rod, and vertically to take up that caused by downward pressure due to the weight of the fly-wheels, etc. Vertical and horizontal adjustment can be obtained, with a two-part bearing, by parting the bushing at an inclination with the direction of both pressures. The inclination is generally made at 45°. The frame-work connected with the crank bearings on horizontal engines is generally part of the engine frame, as in Figs. 279 and 280.

Three-part Bearing.—An example of this form of bearing is shown in Fig. 279, where the horizontal wear is taken up in one direction only, by screwing in the screws A, which move up the adjusting-gibs G against the shaft. The vertical wear is taken up by screwing down the cap C. In this design, as the bearings wear, the shaft will be moved forward and down, and can only be returned to its original position by renewing the babbitt strips. The cap is made to fit into the frame, and is also provided with projections which fit over the outside of

the frame, thus insuring that it will sit squarely upon its journal. To keep the cap (*C*) from being screwed too far and

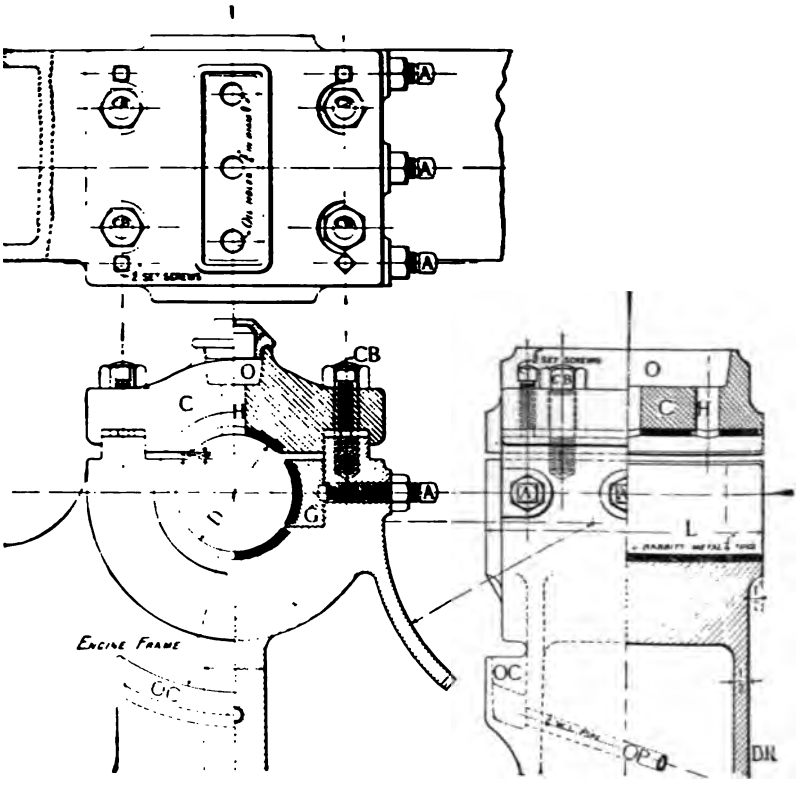


FIG. 279.

clamping the shaft, it is provided with an adjusting screw at each corner.

The lubricator *O* consists of a pocket cast in the cap from which the oil is conveyed to the bearing through the holes *H*. These are filled with cotton to keep the oil from flowing into the bearing too rapidly. This system of lubrication is efficient,

but very wasteful unless the surplus oil flowing from the bearing can be caught and used again. This is done (Fig. 279) by casting a hollow projection *OC* on the frame under the bearing, from which the oil is drained off by the pipe *OP* to the bottom of the engine frame.

Four-part Bearing.—This form of bearing (Fig. 280) is parted on each quarter of the journal, which allows the wear, caused by the thrust and pull on the connecting-rod, to be taken up on either side. This is effected by screwing down the bolts *A*, which pull up the tapered wedges *W*, moving the gibs *C* forward toward the journal. To hold the bolts *A* against turning back, they are provided with a locking arrangement, shown, drawn to an enlarged scale, in Fig. 281. The vertical adjustment is obtained by screwing down the cap-bolts *CB*. The under side of the journal is carried upon a block *LB*, which is allowed to move transversely, thus allowing it to adjust itself to the journal, but is held against moving longitudinally by projections which fit over the raised part *F* on the frame. The top bearing *TB* is held in position by the screws *S*, which also serve to hold the gibs (*G*) in position and keep the cap from being screwed down too tightly on the shaft. The lubricator *O* may be used for semi-liquid grease or by filling it with cotton saturated with oil.

Length of Crank Bearing.—To calculate the length of a bearing it is necessary that we should know the amount and direction of the pressure to which it is subjected. The pressure on the crank-shaft bearings of a horizontal engine is uncertain in amount and direction. We can determine the amount and direction of the resultant pressure caused by the thrust and pull of the connecting-rod and that due to the

weight of the fly-wheels and shaft, but this pressure may be either augmented or relieved by the transmission of the power.

A reliable rule, and one which is generally observed in this country, is to make the length of the bearing equal to TWICE THE DIAMETER OF THE SHAFT.

The Cap.—To relieve the cap as much as possible from the stresses the frame is carried up well around the bearing, and the cap (C) is practically a flat plate. The upward pressure of the cap caused by the angularity of the connecting-rod is found by the formula

$$p = \frac{P}{\sqrt{\quad} - 1},$$

This pressure may be augmented by the gearing which is used to transmit the power, and, to insure that the cap and cap-studs will have sufficient strength under the worst conditions, the value of p should be increased 100%. Then the maximum pressure p' on the cap will be found by the formala

$$p' = \left(\frac{P}{4K^2 - 1}\right)2, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where P = total steam-pressure on the piston;

K = ratio of length of connecting-rod to throw of crank.

The length of connecting-rod is generally made equal to 6 times the throw of crank. The cap is in the condition of a beam on which the load is distributed over its entire surface.

Then the bending moment is $\frac{p'l}{8}$, and the moment of resistance to bending is $\frac{LT^2}{6}f$. Therefore

$$\frac{p'l}{8} = \frac{LT^2}{6}f,$$

from which

$$T = \sqrt{\frac{p' \times l \times 6}{L \times 8 \times f}}, \quad \ldots \quad (3)$$

where L = length of cap;

T = thickness of cap;

p' = total load on cap;

l = distance between cap-studs;

f = strength of the material, which may be taken at 5000 lbs.

Diameter of Studs.—The maximum pressure (p') on the under side of the cap is resisted by the studs CB. Therefore their effective area will be found by the formula

$$\text{Area at bottom of thread} = \frac{p'}{n \times f_t},$$

where n = number of studs;

f_t = strength of material = 5000 lbs. per square inch of area at bottom of threads.

Having found the area at the bottom of the threads, turn to Table No. 8, page 66, from which take the nearest diameter of screw having the required area. The diameter of the adjusting-studs (A) and the set-screws (s) may be made $\frac{5}{8}''$ in diameter when the journal is 6″ or less, and increased $\frac{1}{8}''$ for every inch the journal is increased above 6″ in diameter.

The Gibs.—The height of the gibs (G) should be $\frac{3}{8}$, and their thickness at t should be equal to $\frac{1}{8}$, of the shaft diameter.

Adjusting-wedges.—Instead of using three adjusting wedges and screws, as in Fig. 280, another arrangement is to

use one wedge and one adjusting-screw with two guide pins, as in Fig. 282. In the latter arrangement the wedge supports the gib and is in contact with the frame its entire length. The thickness of the wedges at the top should be $1\frac{1}{2}$ times the diameter of the screw (A) $+ \frac{1}{8}''$, and their width w when

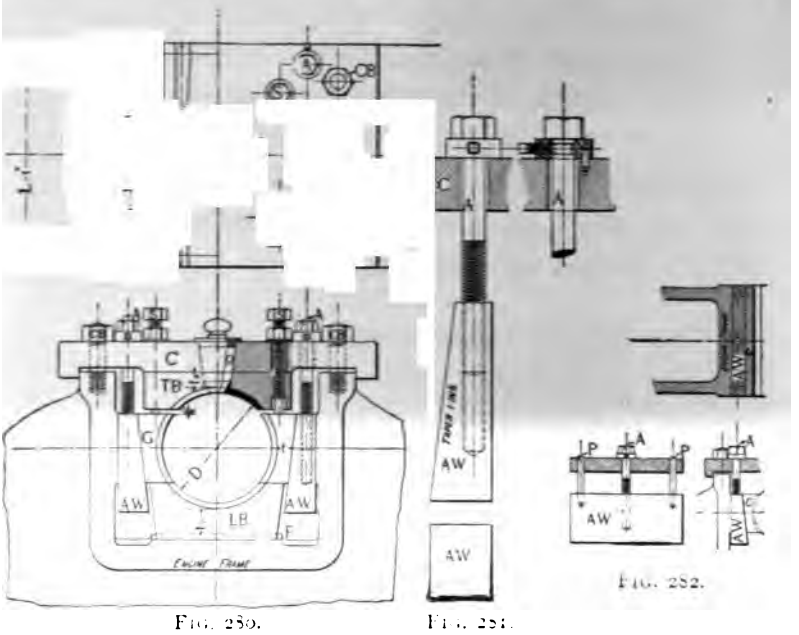


FIG. 280. FIG. 281. FIG. 282.

three are used should not be less than $\frac{1}{8}$ the length of the journal (L). The taper of the wedges may be made from 1 in 6 to 1 in 8. The screw A should be sufficiently long to enter the wedge W a distance equal to its diameter when the wedge is full down.

Top and Bottom Blocks.—The thickness t at the thinnest part of the bottom block should be equal to .23, and that of the top block .15, of the journal diameter.

Exercise 137.—Design a crank-shaft bearing of the form shown in Fig. 279, proportioned for a horizontal steam-engine, having a cylinder 9″ in diameter, stroke 10″, initial steam-pressure 200 lbs. per square inch, and the diameter of the journal (*D*) 4″. The bearing to have a vertical and horizontal adjustment of $\frac{5}{8}$″. Show a HALF ELEVATION, A HALF SECTIONAL ELEVATION, a HALF END VIEW, a HALF SECTIONAL END VIEW, a HALF PLAN, and a HALF SECTIONAL PLAN of the right-hand side. *Scale 8″ to the foot.*

Exercise 138.—Design a crank-shaft bearing of the form shown in Fig. 280, proportioned for a horizontal steam-engine having a cylinder 18″ in diameter, stroke 30″ long, and an initial steam-pressure of 220 lbs. per square inch. The bearing to have a horizontal adjustment of $\frac{1}{4}$″ in either direction and a vertical adjustment of $\frac{3}{8}$″. Make *D* the diameter of the journal 9″.

Show an ELEVATION, PART PLAN, and PART SECTIONAL PLAN, the plane of section passing through the centre of journal. *Scale 4″ to the foot.*

Show also a detail drawing of the adjusting screws and wedges, as in Fig. 281. *Scale 8″ to the foot.*

Exercise 139.—Design a crank-shaft bearing of the form shown in Fig. 280, substituting the adjusting-wedge arrangement shown in Fig. 282. Make the proportions suitable for the conditions given in Exercise 138. *Scale 4″ to the foot.*

Ball Bearings.—This device for reducing friction consists of perfect spheres placed between the journal and the bearing; the balls taking the place of the bush in supporting the shaft, thus substituting rolling for sliding friction. As the bearing areas are only slightly flattened points, the wear will

be con tively rapid; so, to reduce the amount to a minimun e balls and the surfaces upon which they roll are made steel tempered as hard as possible.

The diff rent forms of ball bearings are designated accord- in umber of points that the balls have in contact wi e s aces upon which they roll.

In a -point bearing a line drawn through one of the points in e direction in which the load acts should pass midway betwee oints. Thus the form of bearing shown i ve good results only when the resultant of all th s at an angle of 45°, otherwise the balls will not re on a true axis, but will have a screw motion and therefore a considerable amount of friction. The design shown in I is uitable for a pressure in a vertical direction only. In a point bearing a line drawn through one of the points in the direction in which the pressure acts should pass through a contact-point on the other side of the ball as in Fig. 285), then the balls revolve on a true axis and sliding friction is entirely avoided.

Size of Balls.—Steel balls rolling under pressure do not fail by crushing, their period of usefulness depending upon both speed and pressure. This would seem to indicate that the balls should be as large as possible, thus reducing the number of revolutions in proportion to those of the shaft, and increasing their strength; but there is a practical limit to this owing to the fact that the larger the balls the fewer will be the number, and therefore the fewer the number of bearing points. The bearing would then fail by the balls crushing into the surfaces upon which they roll. There is a great diversity of opinion as to the proper size of ball in relation to

load and speed. The size given in Table No. 37 gives a fair average proportion of the diameter of the ball to the diameter of shaft used in practice for horizontal bearings.

TABLE NO. 37.

Shaft Diam.	Ball Diam.	Crushing Strength of Ball.	Shaft Diam.	Ball Diam.	Crushing Strength of Ball.
$\frac{1}{4}''$	$\frac{3}{16}''$	3 000	$1\frac{1}{4}''$	$\frac{1}{2}''$	20 000
			$1\frac{1}{2}$		
	$\frac{1}{4}$	5 000	$1\frac{3}{4}$		30 000
			2		
	$\frac{5}{16}$	7 000	$2\frac{1}{2}$		40 800
			3		
1	$\frac{3}{8}$	12 000	$3\frac{1}{2}$		50 000
$1\frac{1}{2}$			4	1	60 000

Ball Races.—The thickness (t) of the surfaces upon which the balls roll should not be less than $\frac{1}{2}$ the diameter (d) of the ball, and the width $W = 1\frac{1}{2}$ times the ball diameter. The angles of the grooves are generally made 45°. In every ball race a slight amount of clearance (c) is left between each pair of balls. This is necessary, first to get the balls into place, second to insure the free rolling of the balls. The amount of clearance is generally made from .002 to .004; it will therefore be safe to assume .003 as good practice. Then taking the diameter D_1 of the ball circle $= D + 2t + d = D + 2d$. In Fig. 285 the angle $\frac{1}{2}\theta = \frac{360°}{2n} = \frac{180}{n}$, $\sin \frac{180°}{n} = \frac{d + c}{D_1} = x$.

From a table of sines find the angle θ in degrees corresponding to x. Then

$$\frac{180°}{n} = x, \quad \text{and} \quad n = \frac{180°}{x}. \quad . \quad . \quad . \quad . \quad (4)$$

The number of balls must be within .001 × n of being a whole number; if not, we must increase the diameter D_1. Thus

supposing that formula No 4 gives $n = 20.75$, then we must increase D_1 to get in the next whole number of balls.

Taking $n = 21$, we can find D_1 by the formula

$$D_1 = \frac{d + c}{\sin \frac{180^\circ}{n}} \quad \ldots \ldots \quad (5)$$

Load on Bearings.—As already explained, the life of a bearing is a function of both speed and load. Therefore if

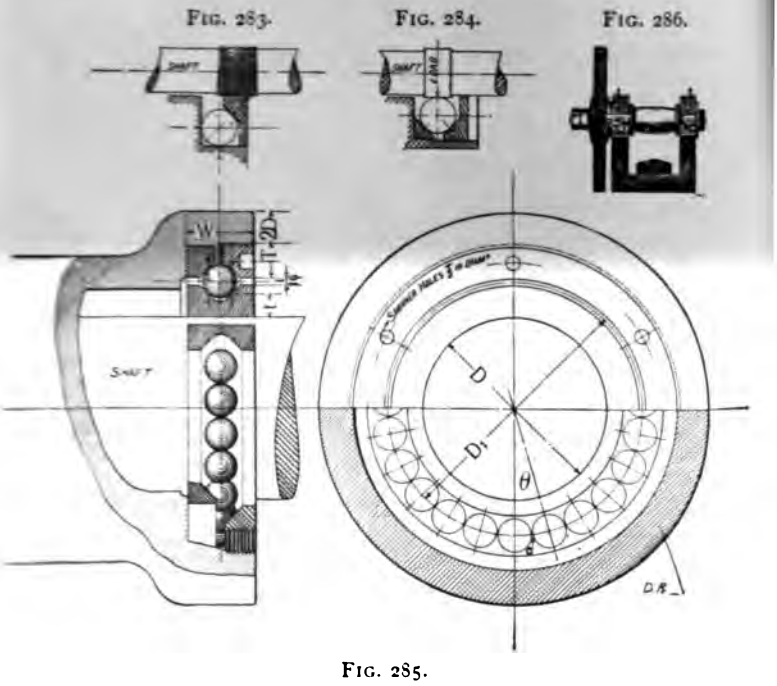


FIG. 285.

the speed is increased, the load must be correspondingly decreased or the life of the bearing will be shortened. Using the proportion of ball to the shaft diameter given in Table

37, the safe load in relation to the speed may be found by the formula

$$L = \frac{f_c \times N}{S}, \quad \ldots\ldots\ldots \quad (6)$$

where L = total load on the bearing;

f_c = strength of ball;

S = speed of the ball races in feet per minute;

N = number of balls carrying the load; in horizontal bearings = $\frac{1}{4}$ of the total number.

Exercise 140.—Design a lathe grinder of the form shown in Fig. 286 provided with four-point ball bearings. Make the emery-wheel 6″ in diameter × ½″ thick, belt drum 1¼″ in diameter and length suitable for a 1½″ belt. The diameter of the shaft (D) = ⅝″.

Show an ELEVATION with one of the bearings partly in section, a HALF END VIEW and a HALF SECTIONAL END VIEW as shown in Fig. 286. *Scale twice full size.*

Thrust Bearings.—The difficulty experienced with the ordinary pivot or thrust bearing, due to the velocity increasing as the distance from the centre, is overcome by the application of balls to this type of bearing. The designs shown in Figs. 287 and 288 are made by the Boston Ball Bearing Co. and may be used on either vertical or horizontal shafts. The balls are held in place by the cage C, the use of which, although tending to increase rather than diminish friction, facilitates the placing and removing of the balls, and by its use the balls can be placed at various distances from the centre of the shaft, thus increasing the time the bearing will run before wearing grooves in the plates PB. In Fig. 288

the balls are arranged in spirals; thus every ball runs on a separate path, and the tendency to wear grooves is reduced to a minimum. The small cages are made in one piece, as in Fig. 287, and the balls are put into position by springing the cage, while in the large cages the top is fastened to the under side, by rivets, after the balls are in position, as in Fig. 288 The cages are made from $\frac{1}{16}''$ to $\frac{1}{8}''$ thick. The thickness (t) should not be less than $\frac{1}{3}$ the diameter of the ball, and the

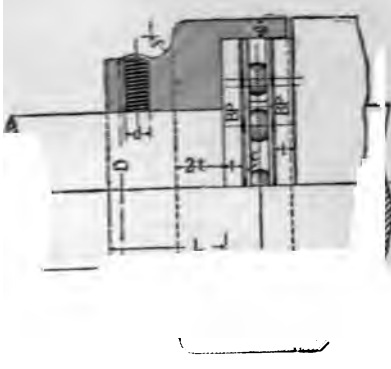


FIG. 288.

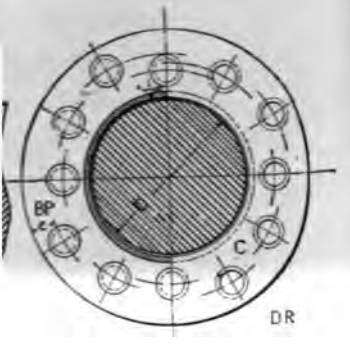


FIG. 287.

distance e should not be less than $\frac{3}{4}$ of the ball diameter. The centres of the ball races are $\frac{1}{4}$ of a ball diameter apart. The hub H is screwed to the shaft by means of one or more set-screws (S), the diameter of which may be made $.2D$, but not greater than $\frac{3}{4}''$. $L = 2t + 3d$, but not less than $\frac{3}{4}D$. $g = .12D$.

$D' = 2D$ when D is less than $2''$, and $= 1.7$ when D is $2''$ or over. The load and speed to which this type of bearing is subjected will determine the number of balls. Taking the size

of ball in proportion to the diameter of the shaft from Table No. 37, then from formula No. 6

$$N = \frac{L \times S}{f_c}.$$

Exercise 141.—Design a thrust bearing of the form shown in Fig. 287 for a 4″ shaft, to carry a load of 760 lbs. and run at a speed of 600 revolutions per minute. *Scale full size.*

Stuffing-boxes.—To prevent leakage, when rods work through the walls of a chamber containing fluid, the rod is passed through a cavity filled with an elastic material which will adjust itself to any irregularities on the surface of the rod. Fig. 289 shows a stuffing-box suitable for a horizontal

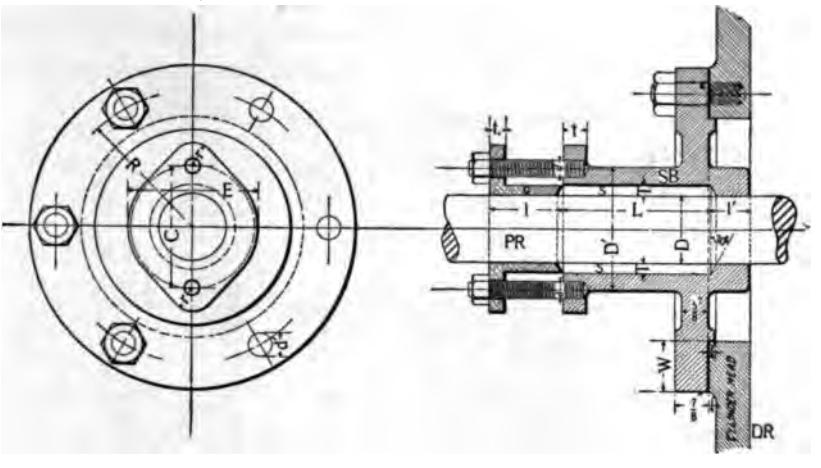


FIG. 289.

steam-engine piston-rod, and Fig. 290 one arranged for a vertical steam-engine piston-rod.

The stuffing-box *SB* may be made a separate piece and bolted to the cylinder-head, as in Fig. 289, or cast with the cylinder-head, as in Fig. 290. Part of the box *SB* is bored

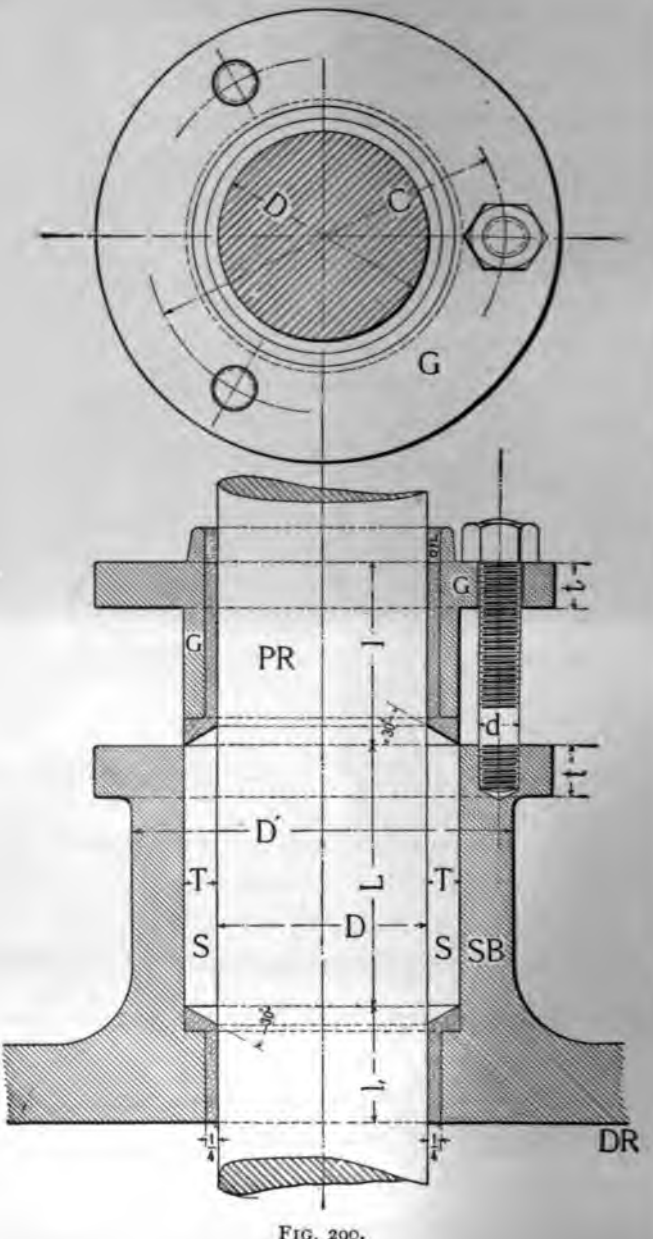


FIG. 290.

larger than the diameter of the piston-rod *PR*, thus leaving a space *S* around the rod which is filled with packing consisting of a fibrous material saturated with oil or tallow. The packing is pressed against the rod by screwing down the gland *G*, which is generally made of brass for rods under 4″ in diameter, as in Fig. 289, and of cast iron lined with brass for the larger rods, as in Fig. 290.

Proportions.—The proportions of the stuffing-box are generally decided by the conditions under which it is used; thus the box is generally made longer for a high than a low pressure. However, under any conditions, the longer the box the longer will the packing last.

The following proportions are suitable for average pressures and speeds, and could be used for high pressure, but would require to be repacked comparatively often:

$L = 2D$ for rods 2″ or less in diameter;
$L = 1\frac{1}{2}D$ for rods between 2″ and 3″ in diameter;
$L = 1\frac{1}{4}D$ " " " 3″ " 4″ " " ;
$L = D + 1''$ for rods over 4″ in diameter;
$l_1 = .75L$; $T = \frac{1}{6}D$ in nearest $\frac{1}{16}$;

$D' = 1.75D + .25$;	$l = .5D + \frac{1}{4}''$;
$d = .2D$;	$r = d + \frac{1}{4}''$;
$C = 1\frac{1}{8}D + 2d$;	$E = D' + \frac{1}{8}''$;
$t = d + \frac{1}{4}''$;	$t_1 = d$;
$R = 2D$;	$d' =$ from $\frac{3}{8}''$ to $\frac{3}{4}''$.

Exercise 142.—Draw a stuffing-box, in which soft packing is to be used, for a horizontal-engine piston-rod (Fig. 289). Make $D = 1\frac{5}{8}''$ in diameter. *Scale full size.*

Exercise 143.—Draw a stuffing-box (Fig. 290), in which

soft packing is to be used, for the H.-P. cylinder of a vertical steam-engine. Make $D = 4''$. Thickness of cylinder-cover $1\frac{1}{4}''$. *Scale 8" to the foot.*

Metallic Packing.—Many designs of metallic packings have been devised to replace the soft packings. One of the most successful is that known as the United States Metallic Packing. A design showing the application of this form of packing suitable for high-pressure steam-engine piston-rods is shown in Fig. 291. [illegible] is known as the "double packing" and is pra[illegible] sets of the ordinary form of packing arranged in t[illegible] n Fig. 291 the back packing is shown in section an[illegible]nt partly in section. The packing consists of bal[illegible]al rings *A*, *B*, and *C* which are cut in halves and [illegible] to the cup *H* by the spiral spring *S*. On the packing [illegible]er the cylinder the spring *S* will be aided by the steam-pres[illegible]ure acting on the follower *F*. The rings *A*, *B*, and *C* are conical, and being forced into the correspondingly shaped cup *H*, the cup-rings close and press against the piston-rod *PR*. The cup *H* rests against the flat face of the ring *R*, which forms a ball-and-socket joint with the outer casing *G* or preventer *P*. As the cup *H* is free to slide on the flat face of the ring *R*, which in turn is free to rock in the casing *G* or *P*, the packing never binds the rod nor constrains it in any way. The packing is prevented from drawing back with the rod (beyond a small movement by the flange on the follower *F* coming in contact with the preventer *P* or *P'*.

Exercise 144.—Draw the arrangement of United States Metallic Packing shown in Fig. 291. *Scale full size.*

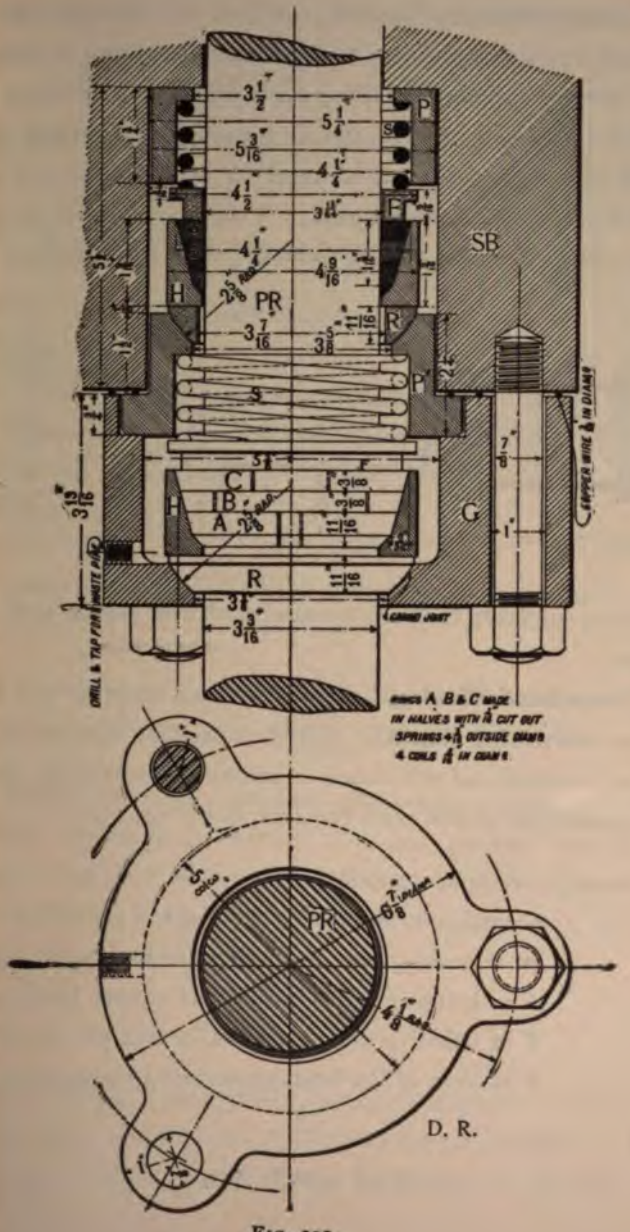


FIG. 291.

Cross-heads and Guides.—When the connecting-rod is inclined toward the direction in which the piston is moving it will exert an upward or downward pressure according to the direction in which the engine is running, and, unless special means are employed, would tend to bend the piston-rod or force it out of its straight path. To prevent such an occurrence the piston-rod end is provided with a cross-head which

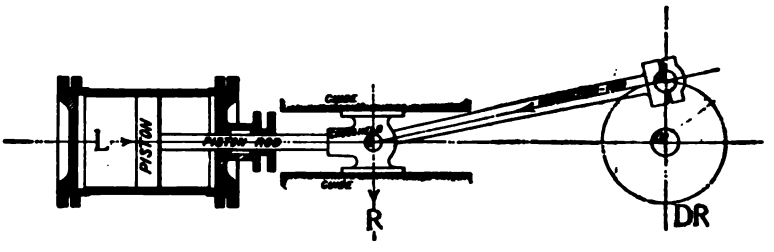


FIG. 292.

slides on surfaces that are parallel with the piston-rod, called guides.

Cross-head Blocks.—Assuming that steam is not cut off before midstroke, then the thrust caused by the obliquity of the connecting-rod will reach a maximum when the crank is nearly at right angles with the line BO (Fig. 292).

Taking L = load on piston;
R = thrust of the connecting-rod;
p = steam-pressure per square inch;
p' = intensity of pressure per square inch;
V = velocity of cross-head in feet per minute;
A = area of the bearing-surface in square inches,—

then

$$L : R : : BO : AO.$$

Therefore

$$R = L \times \frac{AO}{BO} = L \times \frac{AO}{\sqrt{BO^2 - AO^2}}.$$

Taking the length of the crank OA as the unit, then

$$R = L \times \frac{1}{\sqrt{n^2 - 1^2}} = \frac{L}{\sqrt{n^2 - 1}} = \frac{\frac{\pi d^2}{4} \times p}{\sqrt{n^2 - 1}},$$

where n = ratio of connecting-rod to crank.

Pressure on Rubbing-surfaces.—There is great diversity of opinion as to the proper intensity of pressure on the guide-blocks. It varies, according to the different authorities, from 22 to 500 lbs. per square inch. Thurston gives $p' = \frac{40000}{V}$, and that this value in marine and stationary engines may be exceeded to the extent that $p' = 60000 \div V$. Then

$$A = \frac{R \times V}{40000} = \frac{\frac{\pi d^2}{4} \times p \times V}{40000\,(\sqrt{n^2 - 1})}. \quad . \quad . \quad . \quad (7)$$

In many cases of ordinary stationary-engine practice, especially on engines having four-bar guides, the above formula would give a very short block, and as there is generally no difficulty in providing large rubbing-surfaces, we find the areas increased as large as

$$A = \frac{R \times V}{25000}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where V = velocity of piston in feet per minute = (length of stroke × twice the number of revolutions per minute).

The cross-head should always be designed so that the

resultant pressure (R) on the guides will have its point of resistance at the centre of the cross-head rubbing-surfaces, as shown in Fig. 292.

Wrist-pin.—The connecting-rod is attached to the cross-head by the pin CP, Fig. 293. In this form of joint, as the velocity is low and the pressure constantly changing in direction and magnitude, the allowable pressure per square inch is comparatively high, reaching in some designs as much as 1400 lbs. per square inch. Seaton says that the pressure per square inch should never exceed 1200 lbs. per square inch of projected area ($d \times l$).

When the total load on the pin is taken as the maximum load on the piston, i.e., the initial steam-pressure × area of piston, the length of the pin is generally made to equal from d to $1.3d$.

Taking the length $l = d$, then

$$d = \sqrt{\frac{L}{p'}}.$$

When the length of the pin is made equal to $\frac{1}{2}$ of its diameter, then

$$d = \sqrt{\frac{l}{p'}}.$$

Taking the value of $p' = 1200$, then

$$d = \sqrt{L} \div 40 \quad \text{and} \quad l = \sqrt{L} \div 30. \quad . \quad . \quad (10)$$

A pin proportioned to either of the above formulæ will be amply strong to resist bending.

Guide-bars.—When the guiding-surfaces are part of the frame the guides are bored and the bearing-surfaces on the cross-head are turned as in Fig. 297. This arrangement

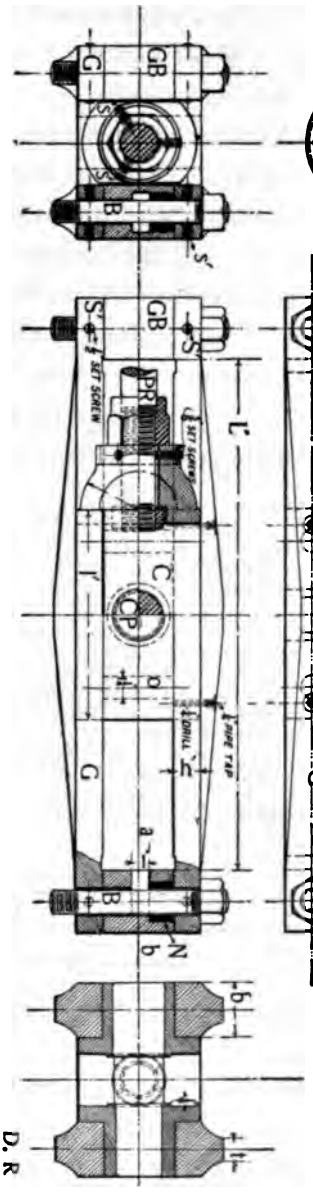


Fig. 293.

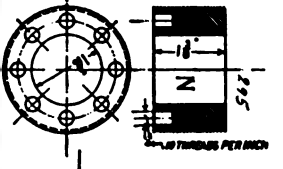


Fig. 295.

D. R

reduces the number of parts, which is always a good point in designing, as it not only decreases the labor but also the liability to error in fitting up. When made separate the bearing-surfaces are flat, and the guide-bars are generally of rectangular (when of steel) or T section (when of cast iron).

To prevent the formation of ridges, due to the travel of the cross-head varying as the wear on the connecting-rod joints is taken up, grooves are cut across the bars over which the ends of the cross-head blocks (*CB*) project at the end of each stroke.

Strength of Guides.—The greatest pressure on the bars occurs when the cross-head is nearly at the centre. Then the bending moment is $=\frac{RL'}{6}$ and the moment of resistance to bending $= fZ$. Where Z is the modulus of section, given in Table No. 29.

The Length of Guide-bars between distance-pieces is = to the stroke + the length of the block + end clearance, which may be made = 1″ at each end.

Four-bar Guide.—The arrangement shown in Fig. 293 is that used on the cycloidal engine (Atlas Engine Works).

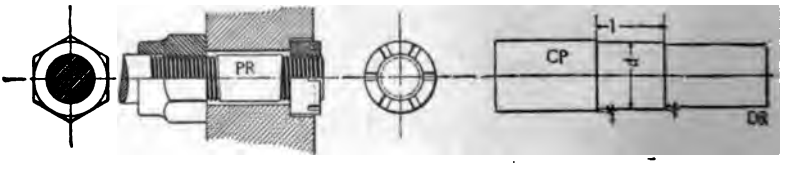


FIG. 294.

With this arrangement the pressure P is equally distributed on each side of the piston-rod, which is guided laterally as well as vertically by the cross-head sliding on the inner surfaces of

the guide-bars G. The piston-rod PR is secured to the cross-head C by the arrangement shown in Fig. 294. To prevent the piston-rod from exerting undue pressure on the stuffing-boxes, should the axis of the piston-rod not coincide with that of the cross-head, the hole through the shank of the cross-head is made larger than the diameter of the piston-rod, which is adjustable and held in position by means of the set-screws S. In this arrangement the breadth b of the bearing-surfaces on each bar is generally made equal to $\frac{1}{4}$ their length.

The area of the bearing-surfaces may be determined by formula No. 8. Then

$$b = \sqrt{\frac{.5A}{4}} \text{ and } l' = \frac{.5A}{b}.$$

The form of guide-bar used in this design may be made of cast iron or steel, and proportioned in the following manner:

Having determined the breadth b and the length L, then calculating for a bar of rectangular section, secured at both ends and loaded at the centre, the height h of the bar at the centre will be found by the equation

$$\frac{RL'}{6} = f\frac{bh^2}{6},$$

from which

$$h = \sqrt{\frac{R \times L' \times 6}{b \times f \times 6}},$$

where f may be taken at 3000 for cast iron, and 6000 for steel.

Take $h' = .75\ h$, then the area of the web will be $= (h \times b) - (h' \times b)$, and taking the thickness t of the web $= .4b$. Then the height of the web at the centre will equal area of web $\div\ 4b$.

The greatest strain on the stud-bolts *B* which secure the guides to the engine-frame is due to screwing up. They may be made = 1½″ in diameter. To allow for any slight inaccuracy of workmanship, the holes through the bars are made $\frac{1}{16}$″ larger than the diameter of the bolts *B*, and the bars are adjusted laterally by the screw *S*. The bars are adjusted vertically by means of the nut *N*, shown in Fig. 295, which is screwed into the guide-bar blocks *GB*. The rubbing-surfaces are lubricated by oil-cups screwed on to the upper guide-bars. The oil is transmitted to the lower bars through the holes *O* on the cross-head.

Exercise 145.—Draw the four-bar guide and cross-head arrangement shown in Fig. 293 suitable for an engine having a cylinder 12″ in diameter × 15″ stroke. Initial steam-pressure 75 lbs. per square inch. Speed 300 revolutions per minute, and a connecting-rod four times the length of the crank. *Scale 4″ to the foot.*

Draw also details of the adjusting-nut *N* and the cross-head pin, and show the arrangement of fastening the piston-rod to the cross-head, taking the diameter of the piston-rod = 2″. *Scale full size.*

Two-bar Guide.—When two guide-bars are used they are arranged either one above and one below the piston-rod (in this case a cross-head of the type shown in Fig. 297 is used) or both guide-bars above the piston as shown in Fig. 296. The latter arrangement is one commonly used in locomotive construction.

The pressure is on the upper guide, *UG*, when the locomotive is running forward, and on the lower guide, *LG*, when running back; and as the engine is generally run forward more

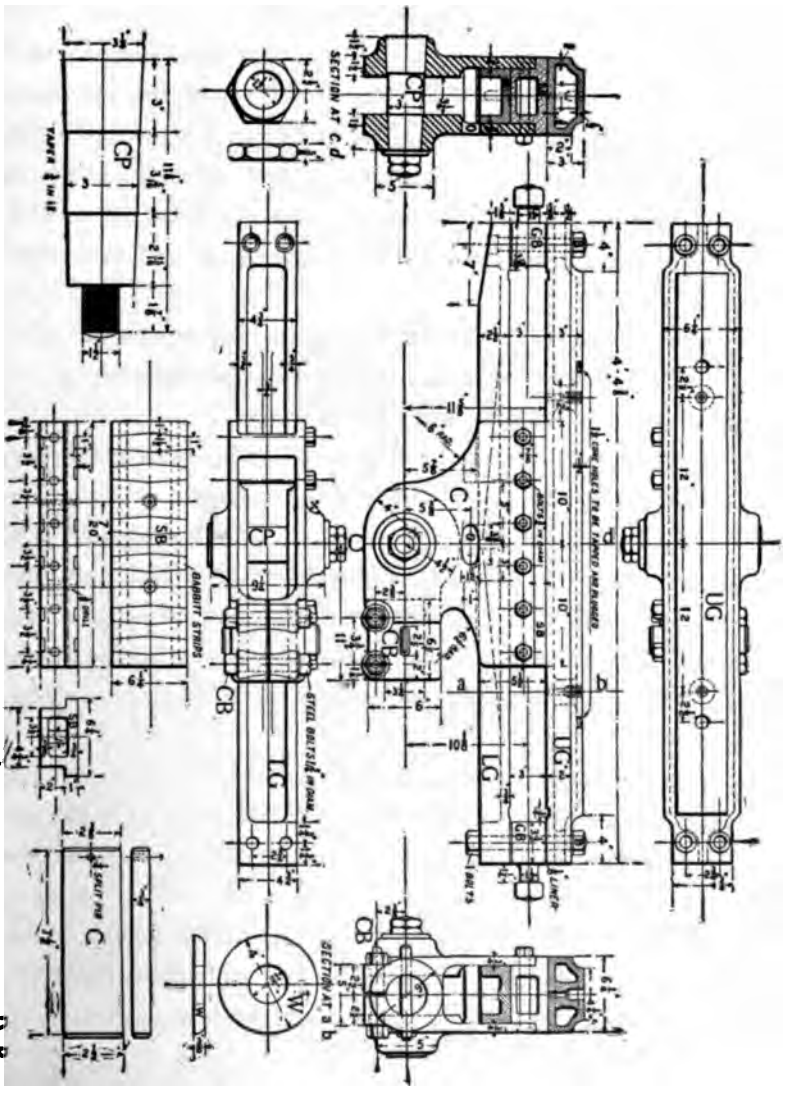

CP
SECTION AT C d
SB
BABBITT STRIPS
CB
LG
C
STEEL BOLTS & NUTS
UG
LINERS
SECTION AT a b

Fig. 206.

D. R.

than back, the bearing-surface on the lower bar may be made smaller than that of the upper.

In this design the cross-head *C* is of cast steel and provided with a brass slide-block *SB* which has strips of babbitt metal top and bottom. To fit and remove the piston-rod easily from the cross-head, the shank is cut and, after the rod is in position, it is gripped by screwing down the bolts *CB* and secured by driving a tapered cotter through it and the cross-head shank.

The hole *O* is to allow for lubricating the cross-head pin.

The guide-blocks *GB* are fastened to the cylinder at one end and to a guide-bar frame at the other.

Exercise 146.—Draw the cross-head and guide-bar arrangement shown in Fig. 296. *Scale 3″ to the foot.*

Also details of the slide-blocks *SB*, guide-block *GB*, cross-head pin *CP*, cotter *C*, and washer *W*. *Scale half size.*

Cross-heads.—Adjustments to take up the wear or for original setting may be accomplished by moving the guide-bars, as in Figs. 293 and 296, or the slide-blocks, as in Fig. 297.

In this design the cross-head *C* is hollowed to receive the connecting-rod end, which works upon the pin *CP*. The pin is of case-hardened steel and is kept from turning by the ¼-inch square-headed screw *K*.

The piston-rod *PR* is screwed into the cross-head and secured by the nut *LN*. The socket into which the rod *PR* is screwed has flat surfaces on the top and bottom to give clearance for the nut *N*.

The diameter at the end of the socket is equal to the dis-

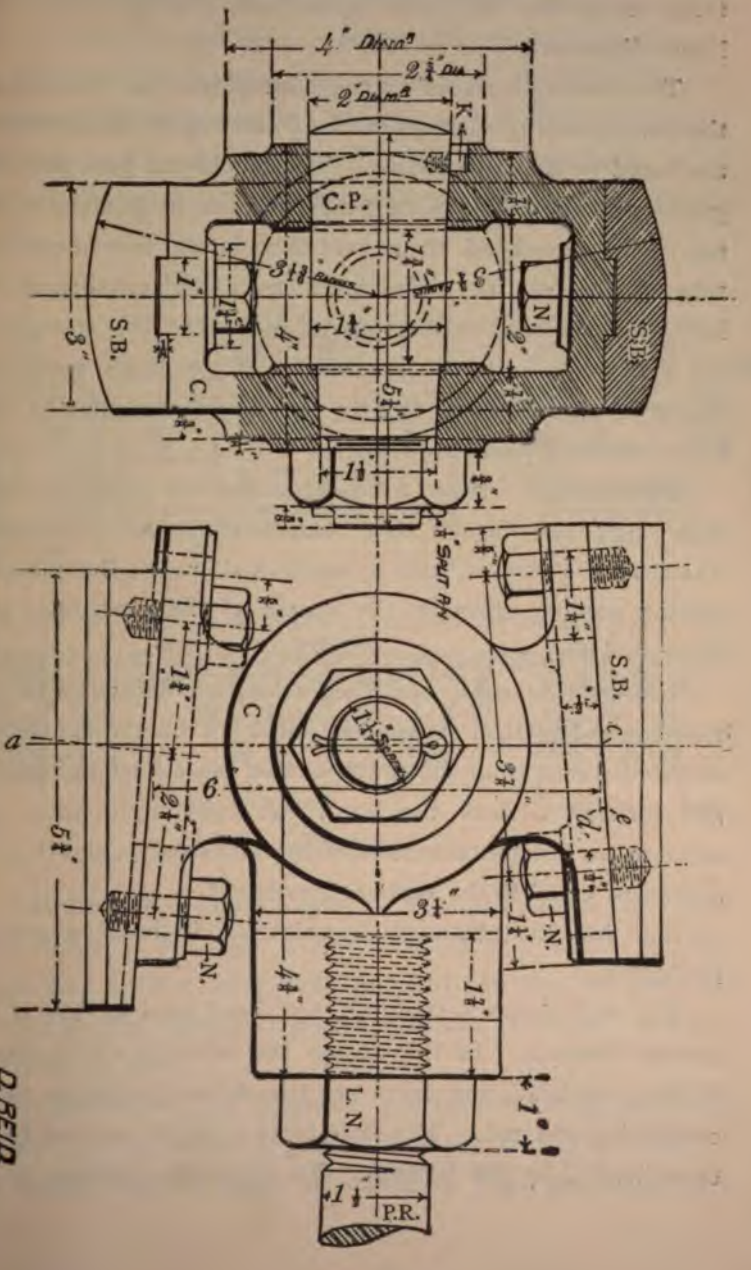


Fig. 297.

D. REID.

tance across the flats, and tapers back $\frac{1}{8}$ of an inch to the larger diameter.

The bearing-surfaces on the slide-blocks are turned, and the corresponding surfaces on the frame, upon which they fit, are bored to the same radius. The blocks are provided with grooves on the under sides, which fit over projections on the top of the cross-head, to prevent their lateral movement. To take up the wear, the slide-blocks move horizontally, on the inclined surfaces upon the bottom of the cross-head, for a distance equal to th of the holes minus the diameter of the stu is horizontal motion they move vertically $\frac{1}{16}$ of a

Exercise 147.—I ead of the form shown in Fig. 297, showing a VATION, END ELEVATION PARTLY IN SECTION, and a SEC IONAL PLAN, the plane of section passing through the centre of the cross-head pin. *Scale full size.*

Construction.—To find the inclination necessary to give the required vertical movement, mark off on the centre line *ab* the distance from the centre of the pin *CP* to the point *C*, and through *C* draw the line *cd* at right angles to *ab* and equal to the horizontal motion of the slide-blocks, and through *d* draw *de* equal to the vertical movement.

The line drawn through the points *ce* will have the required inclination.

Fig. 298 shows a form of cross-head used on the U. S. cruiser Olympia. In this design the wrist-pin *CP* is outside of the cross-head, and there are two bearing-surfaces on the connecting-rod end. The slide-blocks *SB* are secured to the cross-head *C* by the bolts *B*. To allow the removal of the

slide-blocks while the cross-head is in position, one of the projecting lips *L* on each block is removable and held in place by the bolts *B*. To facilitate the removal of the piece *L*, it is provided with set-screws *S*. The piston-rod *PR* is secured to the cross-head by the nut shown in Fig. 67, page 100.

Fig. 299 is an isometric sketch of the complete cross-head.

FIG. 299.

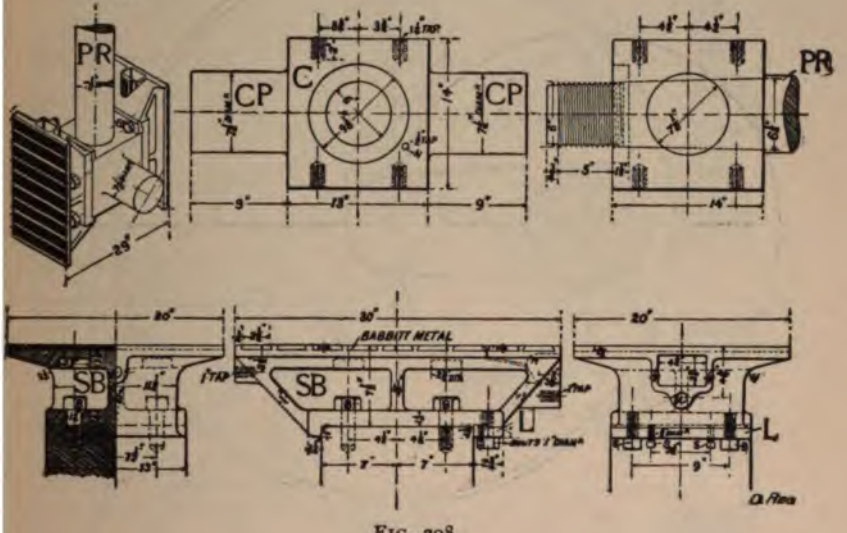


FIG. 298.

Exercise 148.—Draw a general arrangement of the cross-head shown in Fig. 299. Show a FRONT ELEVATION, a HALF PLAN, and a HALF SECTIONAL PLAN of the top, the plane of section passing through the centre of the wrist-pin. *Scale 4 inches to the foot.*

Eccentrics.—The eccentric is a form of crank in which the radius of crank-pin is greater than the sum of the radii of the crank and the shaft, as shown in Fig. 300, where the

crank is shown by dotted lines, and the eccentric by full lines. It is used for converting circular into reciprocating motion. For this purpose its action is identical with that of a crank, and as the eccentric absorbs more power than the crank (owing to the greater leverage at which the friction acts) it is used in preference only where the throw is comparatively

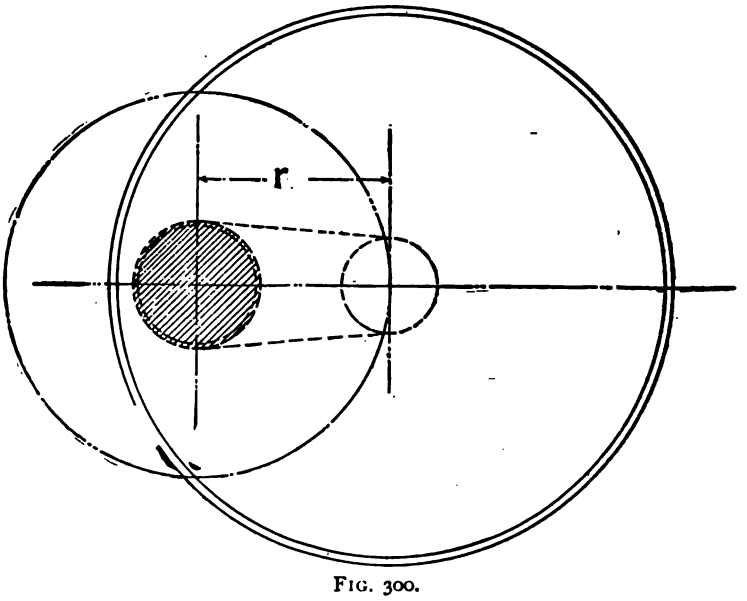


FIG. 300.

short. The eccentricity or throw of the eccentric is the distance r from the centre of the shaft to the centre of the sheave. The stroke of the reciprocating piece worked by the eccentric is equal to twice the throw.

Fig. 301 represents an eccentric used for working the slide-valve of a locomotive engine. The eccentric proper is generally called the sheave or pulley. When it cannot be passed on to position over the end of the shaft, the sheave is

made in two parts, P and P', parted on a line passing through the centre of the shaft and at right angles to the horizontal centre line of the eccentric, and held together by studs. That

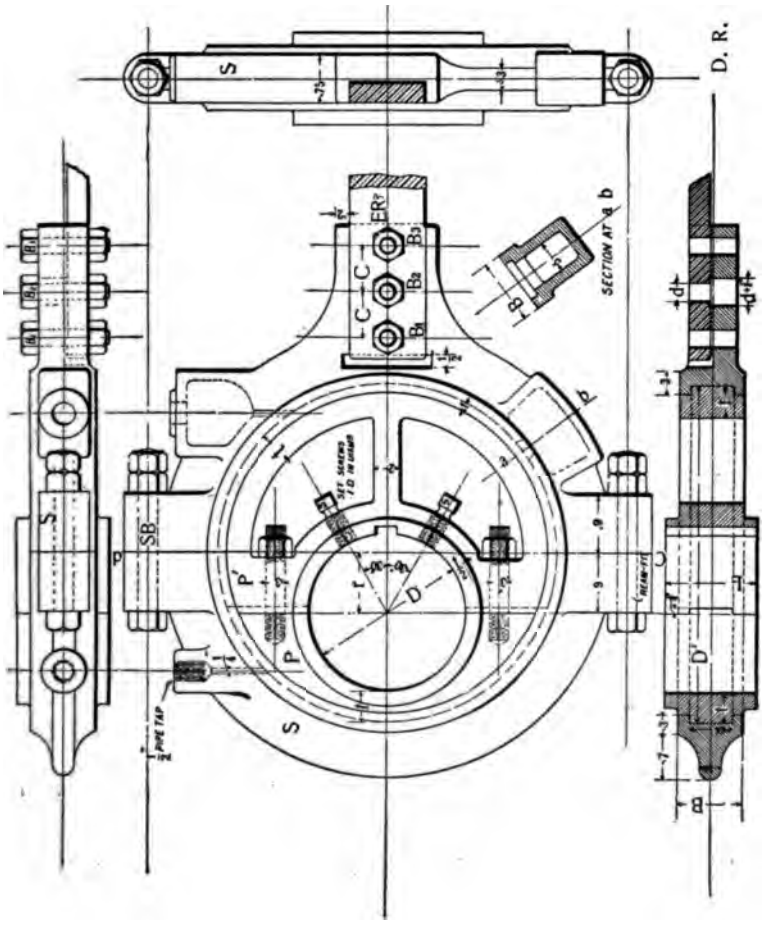


FIG. 301.

the strain may come on the stronger part, P', the key and set-screws used in fastening the sheave to the shaft are placed on that part. The eccentric-rod ER is secured to the strap S by

the b[illegible] B, B_1, B_2. The hole through the strap, for the centre-bolt B_2, is elongated that the rod ER may be adjusted when setting the valve.

Proportions. — The thickness t of the sheave may be $\frac{1}{4}D - \frac{1}{8}''$, with a minimum of $\frac{1}{2}''$. The diameter of the sheave will then $= D + 2r + 2t$. The breadth B of the sheave may be found by the formula

$$B = \frac{L}{D' \times p},$$

where L = load driv[illegible]centric;

D' = diameter[illegible]e;

p = allowable [illegible]r square inch of projected

f_t = area, w[illegible]ve a maximum of 100 lbs.

Thickness of key =

Breadth of key = [illegible]e thickness.

The size of the strap-bolt[illegible] B should be proportioned to resist the load driven by the eccentric.

$$d_1 = \sqrt{\frac{L}{\frac{\pi}{4} \times 2 \times f_t}},$$

where d_1 = diameter at the bottom of the threads;

L = load driven by the eccentric;

f_t = safe strength of bolts, which may be taken at 2000 lbs. per square inch.

The size of the rod-bolts, assuming the load is resisted by the two fitted bolts, may be found by the formula

$$d = \sqrt{\frac{L}{\frac{\pi}{4} \times 2 \times f}}.$$

f may be taken at 3000 for wrought iron. The distance C between centres may be made $= 3d$.

The parts marked in decimals are proportional to B, the breadth.

Exercise 149.—Draw the arrangement of eccentric-sheave and strap shown in Fig. 301, proportioned to carry a load of 2300 lbs., taking the pressure per square inch of projected area = 50 lbs.

Draw the views shown in Fig. 301; also a SECTIONAL END VIEW looking towards the right, the plane of section passing through the eccentric at the line *cd*. Make the eccentric-rod *ER* $3\frac{1}{8}'' \times 1''$. *Scale half size.*



COURSE II.

ELEMENTARY MACHINE DRAWING

INCLUDING

DETAIL WORKING AND ASSEMBLY DRAWINGS, CONVENTIONS FOR DIMENSIONING, INDICATING FINISH, NOTES, BILL OF MATERIAL, TITLE, STYLE OF LETTERING, ETC.

Prerequisites before Beginning Course II. Students must have completed Course I as contained in Reid's Mechanical Drawing, or an equivalent course consisting of Lettering, Geometrical Drawing, Orthographic Projection, Developments, Intersections, Isometrical Drawing and Working Drawing.

ELEMENTARY MACHINE DRAWING.

MINIMUM NUMBER OF PLATES AND MAXIMUM NUMBER OF HOURS ALLOWED TO COMPLETE EACH DIVISION OF THE WORK.

FIRST SEMESTER.

PLATES 1 AND 2. Consisting of *screws*, *bolts*, *rivets* and *riveting* and *bolt fastenings*, must be completed and handed in on or before Friday, October 9, 1908. (26 hours.)

PLATES 3, 4, AND 5. Consisting of *anchor bolt* and *locking devices*, *knuckle* and *cotter joints*, and *shaft couplings*, must be finished according to directions and handed in on or before Friday, November 20, 1908. (36 hours.)

PLATES 6 AND 7. Consisting of *universal shaft coupling* and *pipe couplings*, must be finished and handed in on or before Friday, December 18, 1908. (24 hours.)

PLATES 8 AND 9. Consisting of *post bearing and pedestal bearing*, must be finished and handed in not later than Friday, January 29, 1909. (24 hours.)

Students failing to finish any of the divisions of the work within the time specified, because of excused absences, may make arrangements with the Instructor to work in one or more extra periods.

Students doing more work than is required in the given time will, other things being equal, receive a higher mark.

END OF FIRST SEMESTER.

NOTE.—Registered Freshmen conditioned in Machine Drawing will be required to complete satisfactorily the following plates of this course in addition to the plate shown in Fig. 302: 1, 2, 5, 8, 10, and 13, according to the directions given in the text. Conditioned students must work at least six hours per week. When the above plates are finished, work on the regular Freshman Machine Drawing may be commenced.

SECOND SEMESTER.

PLATE 10. Consisting of *pulley* and *spur gear*, must be completed and handed in on or before Friday, February 19, 1909. (15 hours.)

PLATES 11 AND 12. Consisting of *disk valves* and *globe valve*, must be finished and handed in on or before Friday, March 19, 1909. (21 hours.)

PLATE 13. Consisting of *marine cross-head*, must be finished on or before April 9, 1909. (18 hours.)

PLATES 13 AND 14. These plates must be finished and handed in on or before May 21, 1909. (36 hours.)

Students failing to complete any of the divisions of the work satisfactorily within the time allowed (for good reasons) may make arrangements with the Instructor to work in one or more extra periods.

Students doing more than the required amount of work in the given time will receive a higher mark, other things being equal.

END OF SECOND SEMESTER.

STUDENTS' CONDUCT IN CLASS.

Students will be expected to give strict attention to their ***drawing*** work during the ***full*** time of each drawing period.

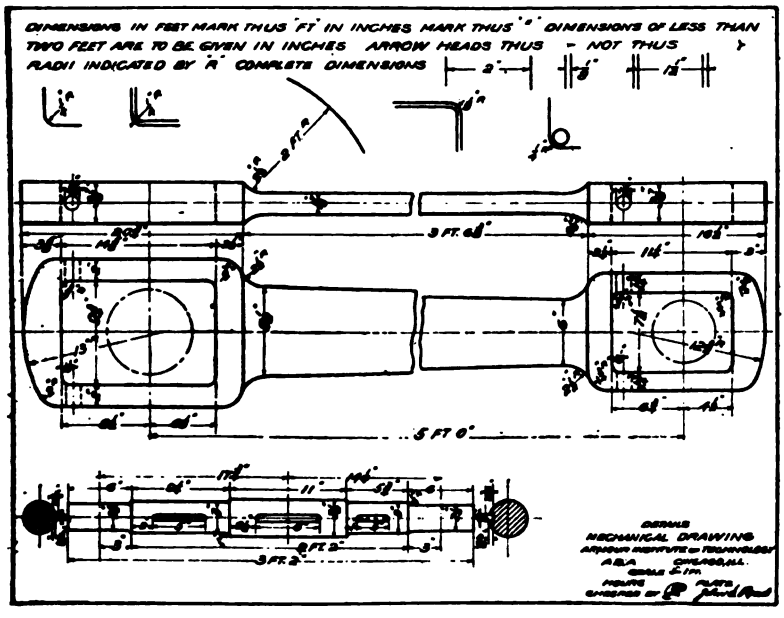


FIG. 302.

Materials and instruments must not be put away until the warning bell rings.

Nothing should be brought to the drawing table that is not needed for the drawing work in hand.

If a student expects to be absent from the class he should endeavor to get excused by the Instructor and make arrangements for making up the work.

A student coming late to class should report at *once* to the instructor, otherwise he will be marked with an unexcused absence.

A report from the Instructor concerning the *deportment* of each student in class is expected by the Dean every two months.

When a student is absent from class through an unforeseen cause, he should at the next regular period fill out an absence blank, giving date and cause of absence, sign it, and hand to the instructor.

The work of all absent periods must be made up by arrangement with the Instructor.

PLATE I. Fig. 35. Exs. 29, page 131. 32, page 142, and 4, page 58. This plate should be laid out similarly to that shown on page 391. See Fig. 304 for standard bill of material. Dimensions for standard title are given in Fig. 304.

Use a 6 H pencil sharpened to a long wedge-shaped point and draw the figures with fine light lines, beginning at the upper left-hand corner of the sheet. In planning the positions of the different exercises take into account the space required for the title and the bill of material. All the exercises should be completed in fine pencil lines and submitted to the Instructor for approval, after which the drawing is to be cleaned and all the given and required lines are to be relined in strong lines with a 4 H pencil sharpened to a long conical point, not too sharp. The dimension lines may now be placed to the best advantage, using a fine sharp line drawn with the 6 H pencil. Care should be taken when locating dimensions *never* to place figures on top of a line or even too near a line when it can be avoided. The dimensions should stand out on the clear paper to avoid con-

usion. Great care should be taken in pencilling in the title and bill of material to see that it is neatly and correctly lettered.

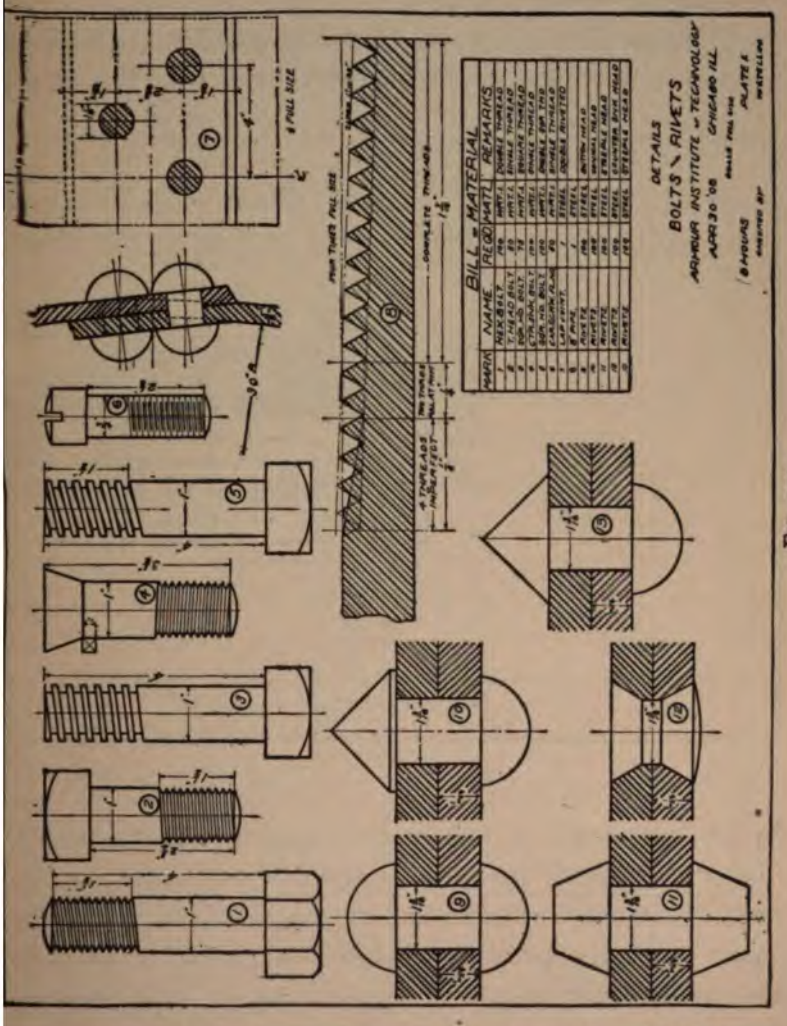


FIG. 303.

The above directions will apply to the pencilling of *all drawings* in this course. Plate 1 will be a finished pencil drawing,

and only the title and bill of material together with the border line are to be inked. The width of the inked border line should be one thirty-second of an inch.

PLATE 2. EXS. 9, 10, 11, 12, and 13, pages 70 to 77. This plate is to be finished in pencil as in Plate 1, and when signed

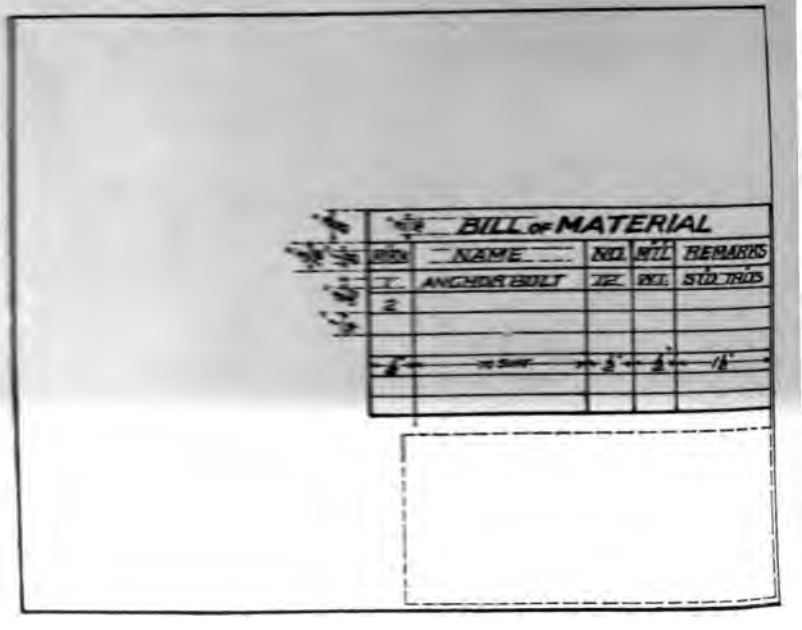


FIG. 304.

by the Instructor it is to be traced on cloth and blue-printed. The proportions for the different styles of bolt-heads that are not U. S. standard will be found on page 75. The plan of the square nut in Ex. 9 and the hexagonal nut in Ex. 10 are to be omitted, but space should be provided for a plan of the *hook* bolt, omitting the nut and washer. In Ex. 13 show the hexagonal nut across corners.

ATE 3. Exs. 17 and 22, pages 85 and 104. This plate ɔe finished in pencil only. Read the text carefully, de-

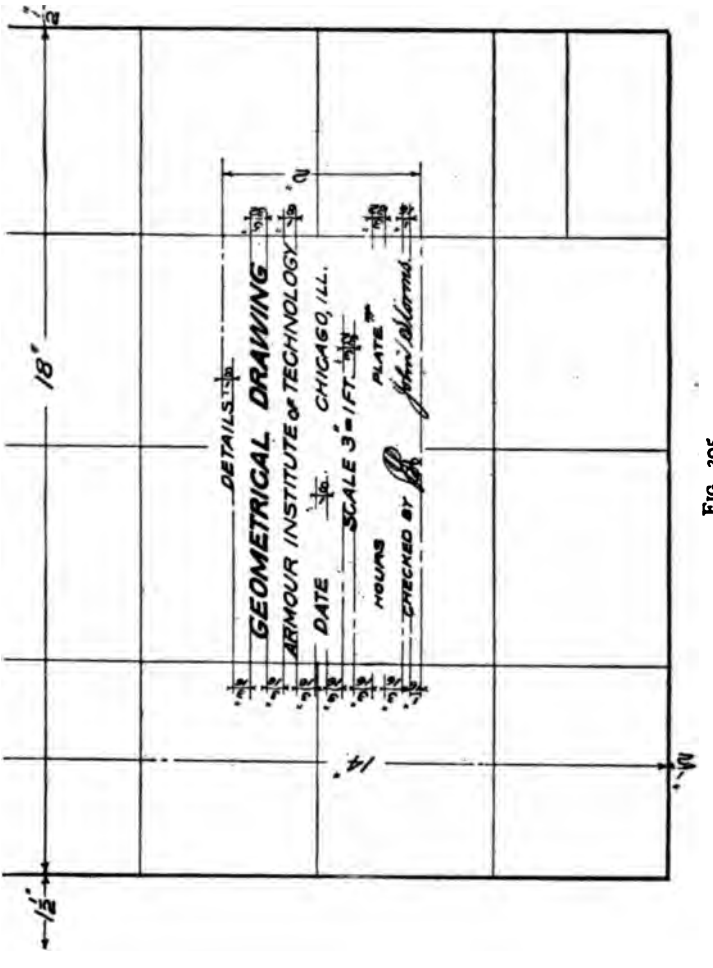


FIG. 305.

ng the different kinds of foundation bolts, viz., the *rag bolt*, *ewis bolt*, and the *anchor bolt*. Study also the require-

ments of the problems and the given data. Before commencing the drawing for Ex. 22 the student should study the text describing the different styles of nut-locking devices. Locate the drawing for Ex. 22 in the upper right-hand corner of the sheet and use only half of the plan shown.

In Ex. 17 assume *D* the diameter of the washer at $6\frac{1}{2}''$ and draw to scale given.

PLATE 4. Exs. 23, 24, and 26, pages 108 and 120. Use only Fig. 84 of Ex. 26. Joints to be made of wrought iron. Finish in pencil.

PLATE 5. Exs. 44 and 50, pages 167 and 178. In Ex. 44 place the longitudinal elevation at left end of sheet and the end elevation to the right of the front elevation and draw also a *plan* above the latter, showing the upper half removed. Make square key $\frac{1}{4}$ of the diameter of the shaft.

In Ex. 50 assume *D* at $17\frac{1}{2}''$. This plate is to be finished in pencil and traced.

PLATE 6. Ex. 53. Page 185. Make detail working drawing. *All* the parts are to be separated and fully dimensioned. Finish in pencil.

PLATE 7. Exs. 56, 58, and 64, pages 191, 194, and 202. In Exs. 56 and 58 make the inside diameter of pipe 6″. Make elevations and sections for all three problems the same as given in Ex. 55. Make finished pencil drawing with title and bill of material as usual.

PLATE 8. Exs. 70 and 109, pages 214 and 304. Use Fig. 225 in the latter exercise. In addition to the two views given in the text-book, show a plan above the main elevation. Make finished pencil drawing and trace on cloth. Blue print.

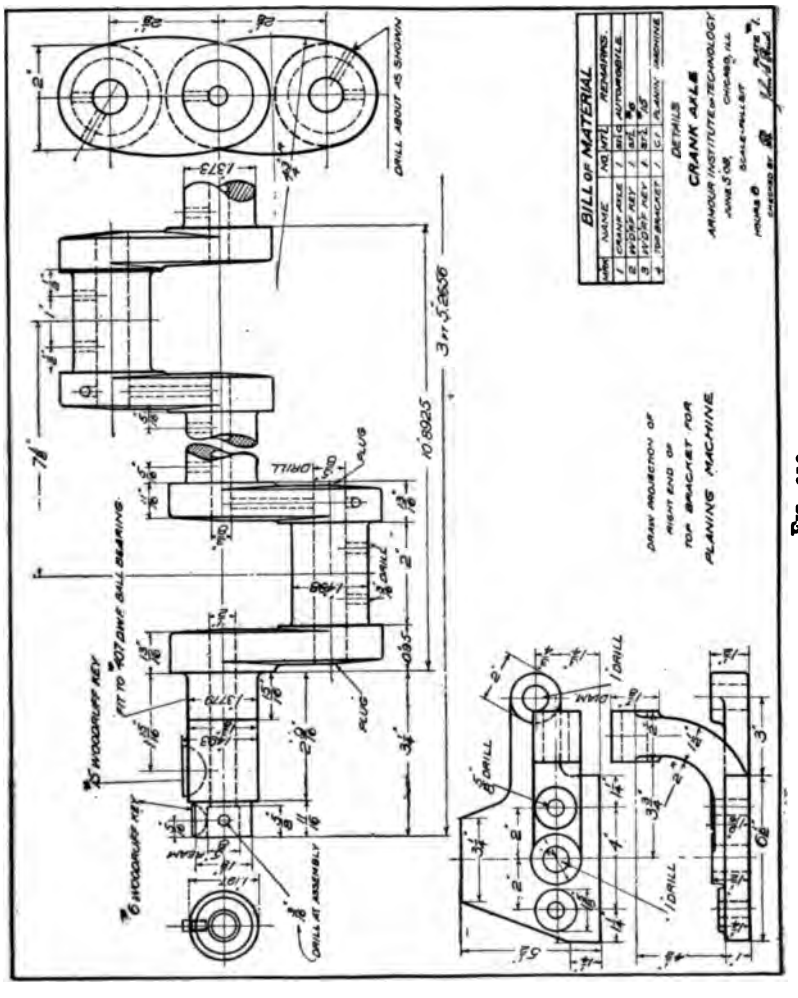

BILL OF MATERIAL
NAME
REMARKS
CRANK AXLE
DETAILS
CRANK AXLE
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
DRAW PROJECTION OF RIGHT END OF TOP BRACKET FOR PLANING MACHINE
DRILL ABOUT AS SHOWN
DRILL AT ASSEMBLY
FIT TO BE A SLIDING FIT
PLUG
DRILL
REAM
WOODRUFF KEY

FIG. 312.

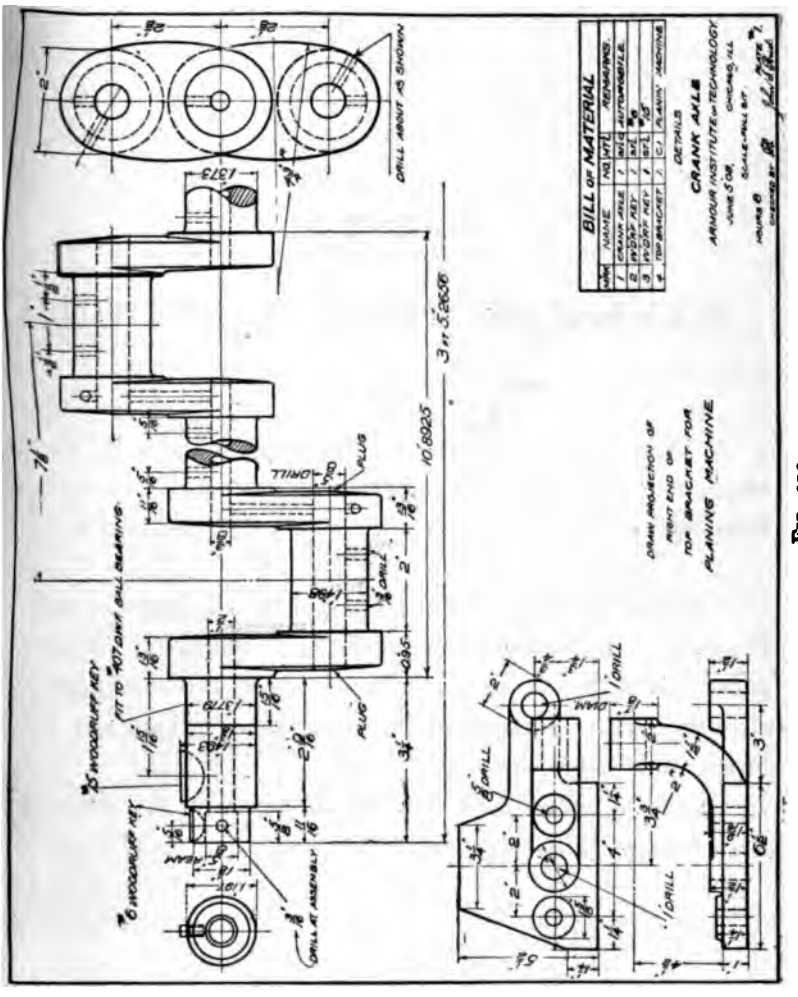

BILL OF MATERIAL
DRILL ABOUT AS SHOWN
DRILL AT ASSEMBLY
PLUG
DRILL
WOODRUFF KEY
FIT TO TOE JOINT BALL BEARING
DRAW PROJECTION OF RIGHT END OF
FOR BRACKET FOR
PLANING MACHINE
CRANK AXLE
DETAILS
ARMOUR INSTITUTE of TECHNOLOGY
CHICAGO, ILL.
SCALE-HALF SIZE
DRAWN BY
CHECKED BY

FIG. 312.

COURSE III.

ELEMENTARY MACHINE DRAWING

FRESHMAN YEAR.

PREREQUISITES.—*Mechanical Drawing.* Course 1, Reid's Mechanical Drawing or its equivalent, consisting of Geometrical Drawing, Orthographic Projection, Intersections, and Developments. 138 hours.

Advanced Mechanical Drawing. Course 2, Reid's Mechanical Drawing or its equivalent, consisting of Isometrical Drawing, Architectural Details, Small Machine Details, Freehand Sketches of small machine parts, Working Drawings, Tracings and Blue Prints. 200 hours.

Student Credit in Class.—Read carefully the directions given on page 389.

COURSE III.

ELEMENTARY MACHINE DRAWING.

INIMUM NUMBER OF PLATES AND MAXIMUM NUMBER OF HOURS ALLOWED TO COMPLETE EACH DIVISION OF THE WORK.

FIRST SEMESTER.

Plates.			Hours.
1–6	due	October 8	26
7	"	October 19	8
8	"	November 5	15
9	"	November 16	8
10	"	December 1	12
11	"	December 17	15
12 and 13	"	January 15	12
14 and 15	"	February 2	15
16	"	February 18	15
17 and 18	"	March 18	20
			146

PLATE 1. *Freehand Lettering*, Fig. 305, page 306.—Use the 4H pencil sharpened to a long conical point, not too sharp.

Fig. 305.

Locate the *lower* point of the first guide-line 12 squares from top and 7 squares from left-hand edge of cross-section pad.

Guide-lines should be sketched lightly with a downward stroke and allowed to remain until letters are approved.

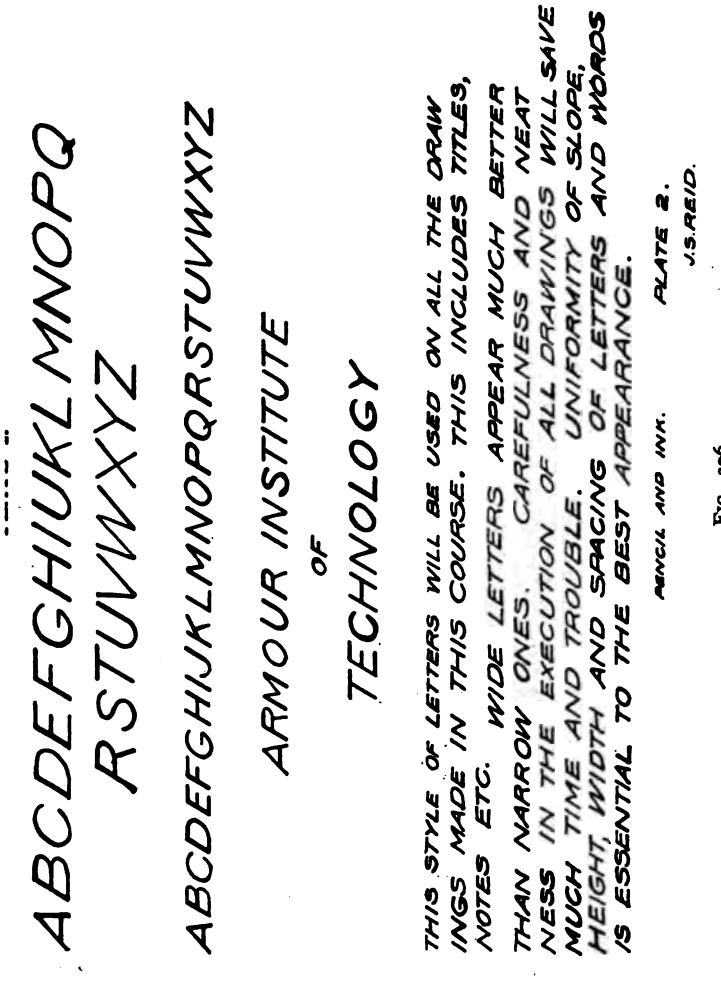


FIG. 306.

After drawing the guide-lines for the *curved* letters analyze the lines of each curved letter, as given on the *chart* on the blackboard before attempting to draw the curves

on the pad. A very close approximation of the first curved letter as it appears on the *chart* should be obtained before attempting to draw the second curved letter.

Do not copy the letters or figures on pages 400 and 404, the correct form and proportions for all the letters and figures must be obtained by a careful study of the *chart.*

The *work* on all the letters and figures must be *strictly freehand.*

Place at the bottom of each plate at the right-hand corner the following information: *Plate* number, *Section* (days and hours), *Time* taken to finish plate, and *Name*, e.g., MON. and WED., 2–4, *Plate* 1. *Time*, 4 *hours*, *Name.* The height of these letters should be *one* square high and all capitals.

PLATE 2. *Freehand* guide lines must be drawn for all letters and figures higher than one square and allowed to remain until letters are approved.

The same care as to *proportion* and *form* should be observed in lettering this plate as in Plate 1.

Be careful to *balance* letters and numbers on all plates so that the same space will appear from both ends of line to edge of pad.

The small letters should be *extended* in width a little beyond the proportion given for the larger letters.

The *open* letters should be spaced closely together and *words* should have a liberal space between them, say 1½ squares.

AND CONSTRUCTION OF THE [illegible]
BE USED ON ALL DRAWINGS THROUGHOUT.

THE STUDENT WILL PROVIDE HIMSELF WITH THE CROSS SECTION PAPER SPECIFIED AND PRACTICE MAKING EACH LETTER UNDER THE DIRECTION OF THE INSTRUCTOR.

OBSERVE CAREFULLY THE FORM AND PROPORTION OF EACH LETTER BEFORE COMMENCING TO MAKE IT.

NO STUDENT WILL BE ALLOWED TO PROCEED WITH THE SUCCEEDING PLATES UNTIL THE ABOVE HAS BEEN COMPLIED WITH, TO THE SATISFACTION OF THE INSTRUCTOR.

AFTER THE PLATES IN LETTERING HAVE BEEN COMPLETED, THE STUDENT WILL BEGIN THE WORK IN MECHANICAL DRAWING.

THE BALL POINTED PENS ARE BEST FOR GIVING UNIFORM WIDTH OF LINE AND WILL BE REQUIRED FOR THIS WORK.

TO BE INKED ONLY.

PLATE *3. HOURS

J. S. REID.

FIG. 307.

PLATE 4.

Pencil three words only of the small letters at first and submit for criticism before going on with the others.

Use *Ball* pen, No. 506, to ink large letters and No. 516 for small letters and figures.

PLATES 3–6.—In the next three *letter plates* the directions for *guide-lines, form, slope, spacing* of letters, and for *width* of small letters should be carefully observed.

PLATE 6.* While a substantial majority of the leading drafting rooms in the United States are in favor of using Gothic Capitals exclusively for notes and titles, there are a number using a combination of Gothic Capitals and Lower Case letters. So it is deemed wise to introduce one plate of Lower Case letters to give the student some knowledge of their form, proportion and construction.

This plate should first be pencilled and after approval, inked. In addition to the "Ball" pen, No. 516, for large letters, the small letters should be inked with Gillott's No. 303. All pens when new should be "exercised" a little before beginning to letter. The form and proportion of these letters as given by the largest letters in Fig. 310, on page 407, should be adhered to as closely as possible.

In general these letters should be made with down strokes of a uniform pressure. The only exceptions are the letters *r*

* All letters and figures should have uniform slope. Letters and figures of one square high should have a full half square slope.

Each *plate* must be signed by Instructor in charge, in pencil before inking and in ink when plate is finished. Plates not so signed will be rejected.

When plates are finished and signed they will be retained by the student until the six plates on *lettering* are completed, when they are to be bound with paper binders and handed to the Instructor.

PLATE 5.

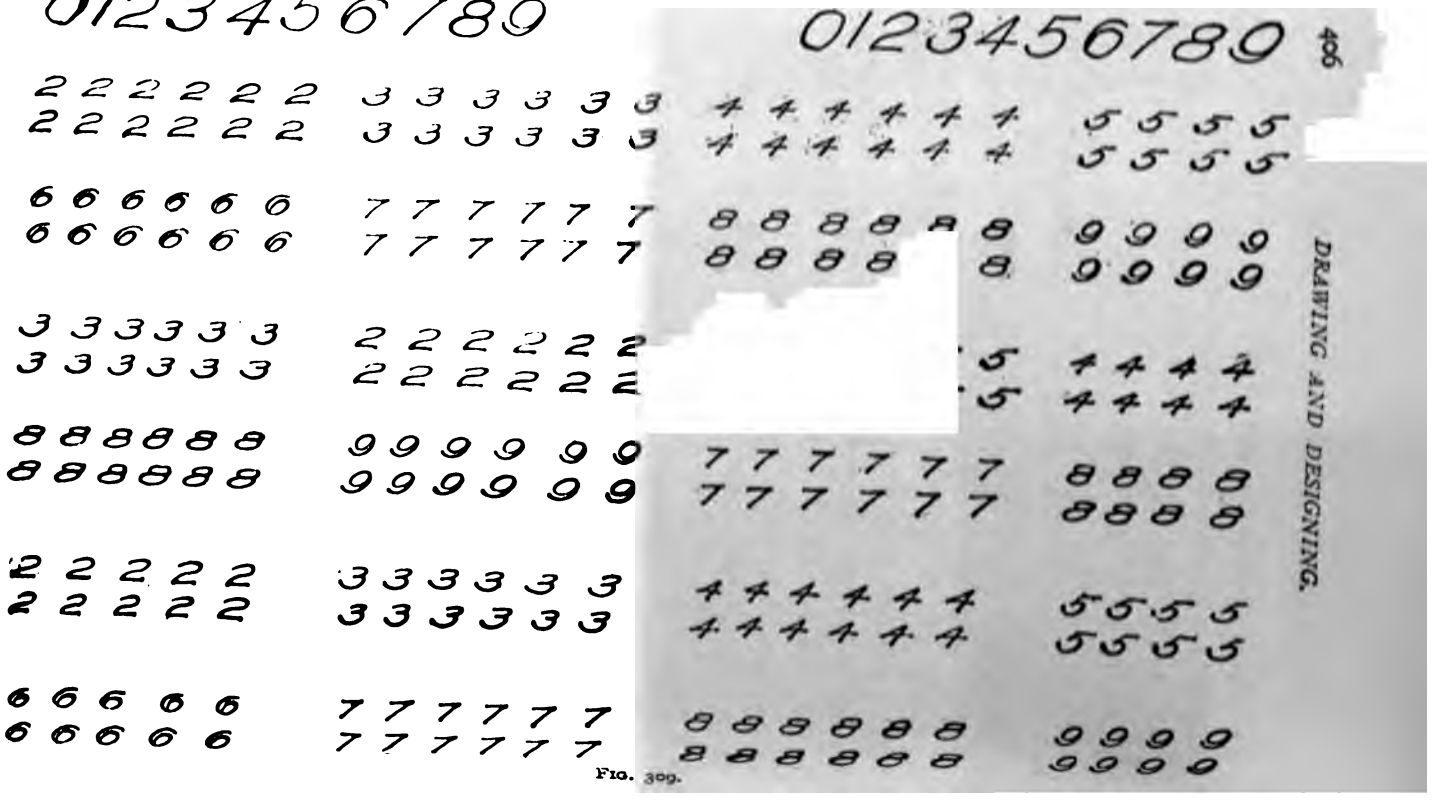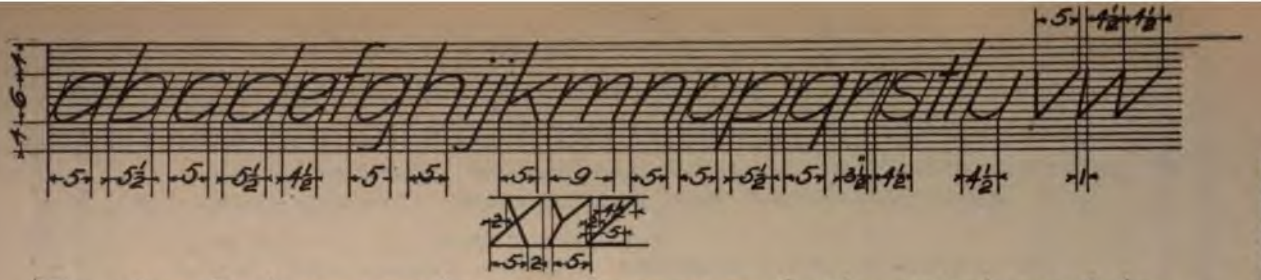


FIG. 309.



The slope for lower case letters is the same as for the capitals, one to two.

In general this style is for everything not covered by style one, and consists of capital letters one eighth of one inch high, and small letters three thirty seconds of an inch high.

All lettering must be freehand. Great care should be taken to make all lettering very neatly and with a maximum of distinctness

Uniformity in slope and height is essential to the best results

That each student may be able to work most effecientlv with out undue fatigue, each should avoid disturbing the other in any way.

FIG. 310.

and *u*. T[illegible] curved part of the *r* may be made with an up stroke
curved only at the top. The *u* is made with two down strokes

PLATE 6A.

THERE IS NOTHING ON THE WHOLE DRAWING SO IMPORTANT AS THE DIMENSIONS.

OVERHEAD CAREFULLY HOW THEY ARE MADE ON THE BLUE PRINT.

IN PLACING DIMENSIONS ON A DRAWING OBSERVE THESE GENERAL RULES:-

1. DIMENSIONS SHOULD NOT BE PLACED SO [illegible] OSE TO OTHER LINES AS TO IMPAIR CLEARN[illegible]S.
2. NEVER PLACE DIMENSIONS ON GEN[illegible]S.
3. DIMENSIONS SHOULD BE BASED [illegible]NTER LINES OR FINISHED SURFACES.
4. ALL DIMENSIONS SHOULD READ FROM THE BOTTOM AND RIGHT HAND SIDE OF A DRAWING.
5. DIMENSIONS ON A DRAWING SHOULD INDICATE THE FULL DIMENSIONS OF THE OBJECT, INDEPENDENT OF THE SCALE OF THE DRAWING.
6. WHEN OVERALL DIMENSIONS ARE REQUIRED THEY SHOULD ALWAYS BE PLACED OUTSIDE OF SUB-DIMENSIONS.
7. SECTION LINING SHOULD GENERALLY BE MADE AT 45 DEG. AREES.

TO BE INKED ONLY.

and the bottom curve filled in with a stroke to the right and upwa
The *m*, *n*, and *h* should be formed with nearly sharp upper curv

This plate will have to be repeated until the desired results have been obtained.

PLATE 6*A*, Fig. 311. This is an extra lettering plate for those students who may finish the required plates ahead of time. The extra plate will increase the grade mark.

PLATE 7.

Bolts and rivets (Plate 1, page 390). Read carefully the directions given in connection with Plate 1 on page 390.

PLATE 8.

Bolt fastenings (Plate 2, page 392).

PLATE 9.

Shaft and Pipe couplings, Exs. 44 and 58, pages 167 and 194 respectively.

PLATE 10.

Post hanger and oil cup (Plate 8, page 394).

PLATE 11.

Marine crosshead (Plate 13, page 395).

Plate 12.

Problem 1. Make a careful freehand sketch of either the 1½" drop hanger or the post hanger to be found in the drafting room. Use orthographic projection and sketch on 8" × 10" cross-section pad with 4 H pencil. Use only one side of the paper. Sketch three views of body and detail of link. All dimensions notes, title, and finish mar[illegible] be neatly placed on sketch. Begin the sketch by drawing all [illegible]e center lines for the front and end elevations and the plan. Sketch center line for detail of link. Make size of sketch to suit size of paper. Lines should be sketched very lightly at first and when sketch is approved the lines may be strengthened. Put on *all* dimension lines before measuring the object. Sufficient dimensions must be placed on sketch to enable the draftsman to make a working drawing for the pattern maker without having recourse to the object, after the drawing is commenced. Sketch must be signed by the Instructor. Callipers may be had at the office.

Plate 13.

Problem 1. Make working drawing from the drop or post hanger, as the case may be, from the sketch made in Plate 8. Make finished pencil drawing scale 6" = 1 foot. See Fig. 156, page 220, and Fig. 159, page 224, for similar hangers.

Plate 14.

Problem 1. Make freehand sketch for water cylinder of steam pump without the adjacent parts. Make longitudinal section through the center of cylinder, plan of top, half cross-

section, and half end view combined, also cross-section through supply and delivery ports. Scale about half size. Large circles may be drawn with compass. Follow directions given for Plate 12.

Problem 2. Make sketch of upper valve seat of water cylinder. Show top and bottom views and turned section through center. Put on dimensions, notes, finish marks, and title on all freehand sketches unless otherwise directed.

Problem 3. Make freehand sketch of valve chest cover for water cylinder, showing top and bottom views and turned section through center.

Problem 4. Make freehand sketches of the following remaining details of the water cylinder of steam pump:

1. Piston and rod.
2. Stuffing-box.
3. Gland.
4. Ring.
5. Air chamber.
6. Valve seat.
7. Valve disk.
8. Valve spring.
9. Valve cover.
10. Valve spindle.

1, 2, 3, 4, and 5 may be sketched on one sheet, using another sheet for 6, 7, 8, 9, and 10. Make all necessary views and complete as before. See sample sketch, Fig. 313, page 412.

PLATE 15.

Problem 1. Make assembly drawing for water cylinder for Steam Pump from sketches in Plate 14. Put on the principal dimensions, notes, title, and bill of material. Make finished pencil drawing and trace on cloth. Scale 8″ = 1 foot.

AIR CHAMBER.
ACROSS FLATS
S.B. RING
STUFFING BOX.
LONG.L SECTION OF PISTON.
DETAILS
WATER CYL. FOR ST. PUMP
FINISH ALL OVER
S.B GLAND
DRILL

FIG. 313.

PLATE 16

Problem 1. Make freehand sketches of either globe or safety valve in detail on one sheet as follows:

1. Globe valve body, longitudinal section, half cross-section and half end view combined.

2. Valve stuffing box and bonnet, half elevation and half section combined, and plan.

Problem 2.

3. Valve spindle, one view.

4. Valve spindle nut, half elevation, half section and plan.

5. Hand wheel, half elevation, half section, and plan.

6. Valve, half elevation, half section, and plan.

7. Valve nut, half elevation, half section, and plan.

8. Stuffing box cap, half elevation, half section, and plan.

Problem 1.

1. Safety valve casing, half elevation, half section combined, and plan.

2. Valve seat, half elevation, half section combined, and plan

3. Ring, half elevation, half section combined, and plan.

Problem 2.

4. Valve spring, elevation and plan.

5. Valve spindle, one view.

6. Adjusting screw, two views.

7. Valve disk, two views.

8. Spring washer, two views.

9. Adjustable screw bushing, two views.

10. Lock nut, two views.

PLATE 17.

Problem 1. Make detail assembly working drawing of either the globe or safety valve from sketches made in Plate 16.

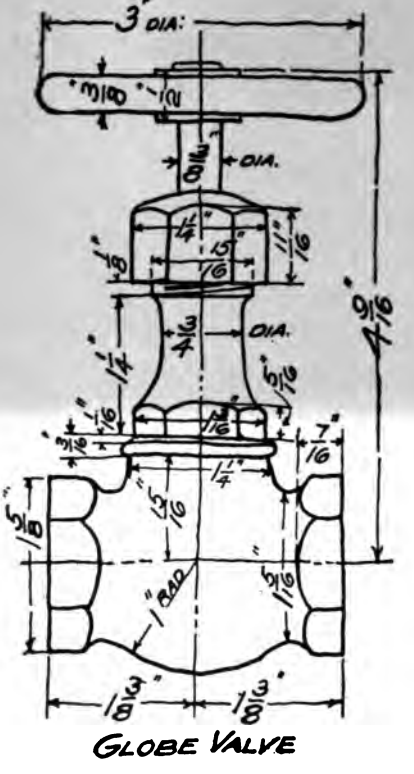


FIG. 314.

This drawing must be fully dimensioned and contain notes, title, and bill of material. Finish in pencil only. Scale to suit.

PLATE 18.

Problem 1. Make isometrical drawing of layout of piping shown in Fig. 315.

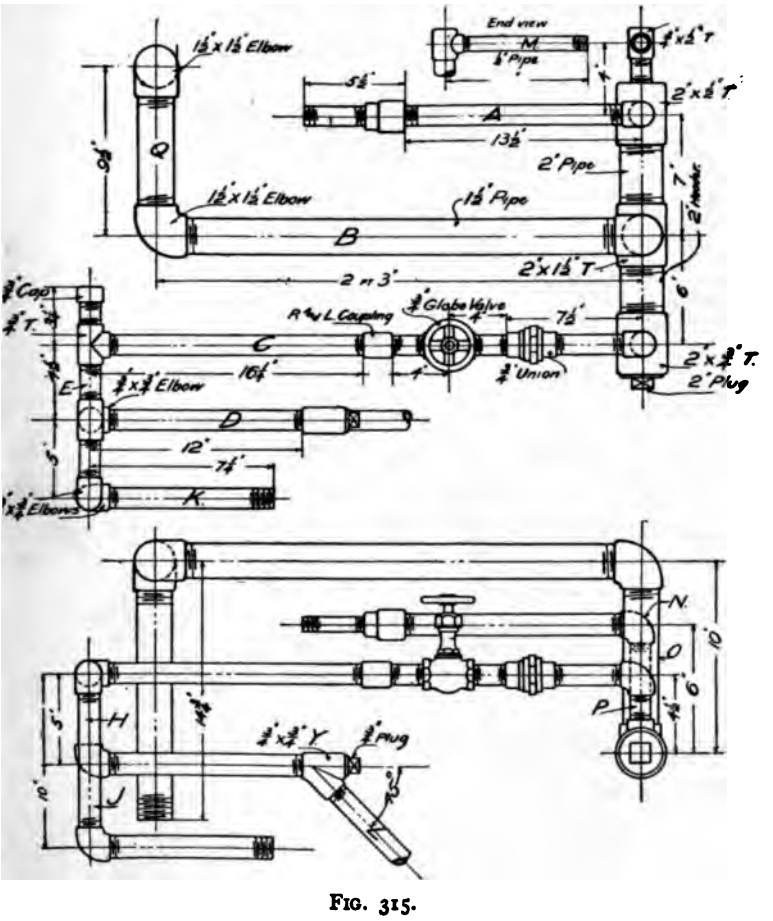


FIG. 315.

Samples of couplings and piping will be found in the drafting room. See Fig. 314 for sketch of ¾" globe valve. Diameter

and thickness of pipes, length of screw, etc., can be obtained from Table 7, page 57.

Fig. 316, page 416, shows the isometric layout of the center lines of the piping. Arrange on sheet to best advantage, allowing space for title and bill of material. Letter all notes very carefully using lower case letters and Gothic caps. Make caps one

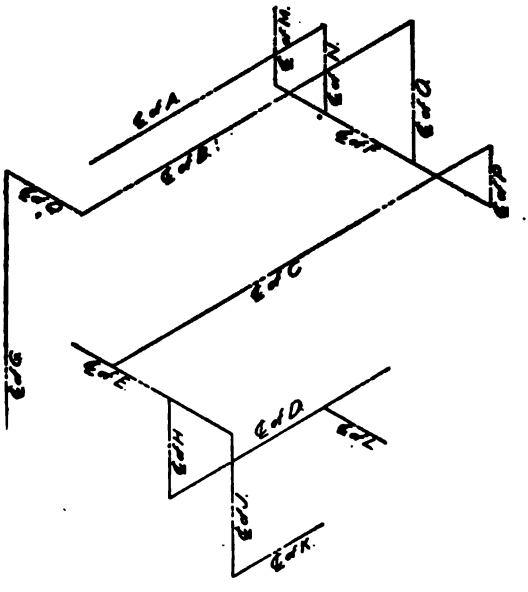


FIG. 316

third higher than small letters. Scale 4″=1 foot. Finish in pencil and trace.

PLATE 15.

If any student has time and desires to increase his mark, he may make as an extra plate an assembly detail working drawing of a chain oiling bearing similar to that shown in Fig. 161, page 226, except that it shall be designed for a 1⅛″ shaft to

either the drop or post hanger of Plate 9. Scale = full size. sh in pencil and trace on cloth, or a design for a donkey

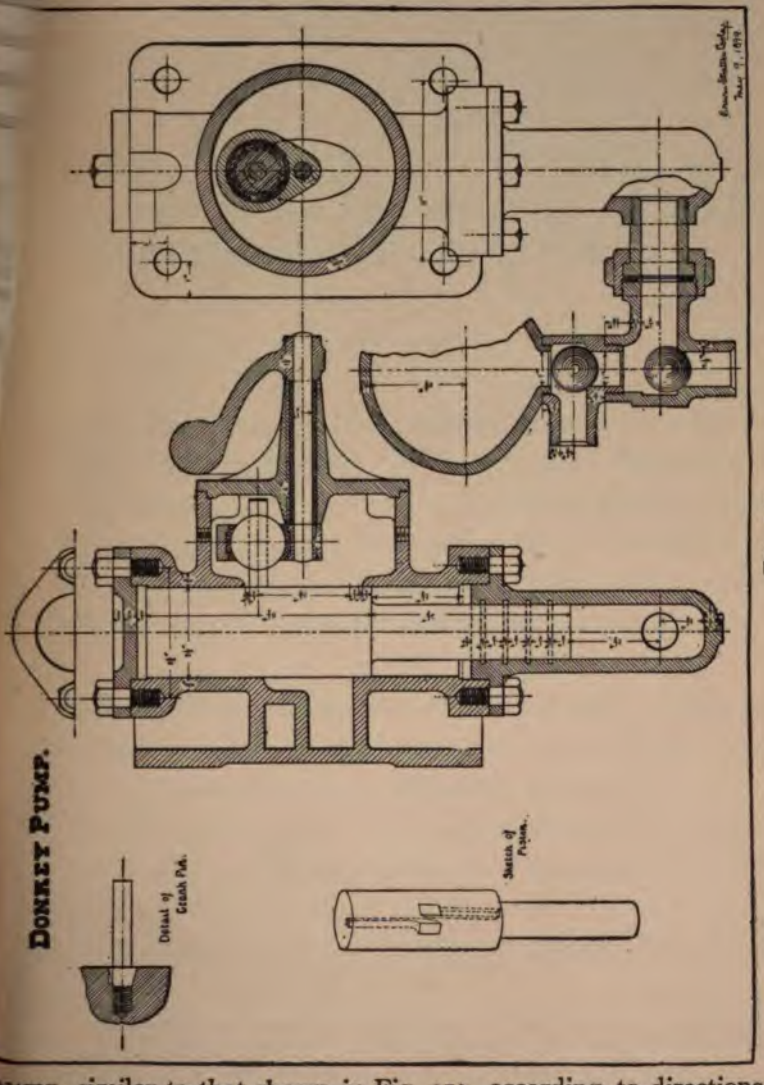


FIG. 317.

Pump, similar to that shown in Fig. 317, according to directions and data to be given by the Instructor.

PRESENT PRACTICE IN DRAFTING ROOM CONVENTIONS AND METHODS IN MAKING PRACTICAL WORKING DRAWINGS.

SUMMARY REPORT OF AN INVESTIGATION MADE BY THE WRITER WITH THE AUTHORITY OF THE ARMOUR INSTITUTE OF TECHNOLOGY, CHICAGO, ILL., INTO THE PRESENT PRACTICE OF THE LEADING DRAFTSMEN IN THE UNITED STATES, IN THE USE OF STANDARD CONVENTIONS AND METHODS WHEN MAKING COMMERCIAL WORKING DRAWINGS.

A circular letter accompanied by a list of thirty-five questions was submitted to two hundred leading firms in the United States, embracing nearly all kinds of engineering practice.

The returns have been exceedingly gratifying, and especially so has been the spirit with which the "Questions" have been received and answered.

Many requests have been received from chief draftsmen for a copy of the returns.

The questions submitted and the answers received are given somewhat in detail below.

Q. 1. Do you place complete information for the shop on the *pencil drawing*, such as all dimensions, notes, title, bill of material, scale, etc.?

Complete information is placed on drawing before tracing.	57
Complete information is placed on tracing only	42
Principal dimensions and title only on pencil drawing	2
Draw directly on bond paper	10
Did not answer this question	10
Sometimes	7

Reasons given for making the pencil drawing complete:

To arrange notes. To save ime. The tracing is not usually made by the draftsman who makes the pencil drawing.

Q. 2. Do you ever ink the pencil drawing?

Never ink the pencil drawing	91
Generally ink the pencil drawing	7
Sometimes ink the pencil drawing	8
Sometimes ink the pencil drawing and shellac it for shop use.	1
Use bond paper	10
Make pencil drawings on dull side of tracing cloth	2
Ink center lines of assembly drawing	1
Ink center lines of pencil drawings in red	2

Q. 3. Do you trace on cloth and blue print?

Always trace on cloth and blue print	102
Blue print from bond paper	10
Blue print from bond paper occasionally	1
Sometimes make "Vandyke" prints for shop use	1
Sometimes use paper drawings in shop for jigs and fixtures.	1

Q. 4. Do you use blue prints entirely in the shop?

Use blue prints altogether in shop	105
Sometimes use pencil drawings or sketch	21



Sometimes use sketches made with copying ink	1
Sometimes use prints from "Vandyke"	1
Use white prints mounted on cardboard and varnished	1
Use blue prints mounted on cardboard	1
Use sketches for rush work	1

Q. 5. When tracing do you use uniform wide object lines? Ever use shade lines?

Use uniform, thick object lines. Never use shade lines	100
Sometimes use shade lines	21
Use shade lines on small details	5
Always use shade lines	14
Experts in the use of shade lines may do so to make drawings clear	1
Shade rounded parts	1

Q. 6. What kind of a center line do you use?

Long dash, very narrow, and dot, thus: ——— · ———	42
Long dash and two dots, ——— · · ———	29
Very fine continuous line, ———————	19
Very fine dash line, long dashes, — — — —	8
Long dash and dot in red, ——— ———	3
Continuous fine red line, ——— ———	8
Long dash and three dots, ——— ———	1
Long dash and two dots, thus: ——— · · ———	1

Q. 7. What kind of dimension line do y

Continuous fine line, broken only for dim	52
Fine long dash line, ——— ———	32
Fine long dash line and dot, ——— ———	13
Fine continuous red line, ———————	8
Fine continuous blue line, ———————	4
Fine continuous green line, ———————	1

Sloping? Ver

height? or ca

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10. Do you use a border line on drawings?

Always use border lines	97
Never use border lines	13
Use border lines on foundation plans, to send out	1
No border lines on detail drawings	1
Intend to discontinue the use of border lines	1
Border lines used only on design drawings	1
Only on drawings to be mounted on cardboard	1
Only used for trimming blue print	2
On assembly drawings only	1

Width of margins reported: 1″, ½″, ⅜″, ¼″, and ⅛″.

11. When hatch-lining sections, do you use uniform or symbolic hatch lines?

Standard symbolic lines	59
Uniform hatch lines for all materials	44
Shade section part with 4H pencil and note name of material	4
Symbolic hatch lines and add name of material	3
Uniform hatch lines for metal only	1
Uniform on details, symbolic on assembly drawings	5
Pencil hatch on tracings and note material other than cast iron	1
Uniform hatch lines, sometimes solid shading	1
No uniform system	1
Sections tinted with water colors representing the metals	1

12. Is the pencil drawing preserved? Is the tracing stored or do you make "Vandyke" prints for storing away?

Store tracings only	96
Pencil drawings preserved for a time	30
Pencil drawings preserved	13
White prints made and bound for reference	1
Tracings kept in office for reference, blue prints stored	9
"Vandyke" prints stored	1

Use "Vandyke" as substitute for tracing 2
Arrangement drawings preserved, detail drawings destroyed after job is completed. Pencil drawings used for gasket paper 1
Original pencil drawing inked and stored 1
Assembly drawings and layouts preserved 4
Patent office drawings preserved 1
Tried "Vandyke" but found it unserviceable, tearing easily. 1

Q. 13. Do you use 6H grade of pencil for pencil drawings or what?

6H 73
4H, mostly for figures and letters 52
5H 16
Ranging from 2H to 8H 53

Q. 14. Do you use plain orthographic projection for free-hand sketches? Ever use perspective or isometrical drawing for sketches?

Plane orthographic 3d angle projection 99
Isometrical drawing for sketches 25
Perspective for sketches 1
Isometric for piping layouts and similar work 8
Perspective and isometric for catalogue work 2
Isometric sometimes 6
Never use free-hand sketches 6

One says, "When we run into other than orthographic, men are too timid and not sure of themselves. In perspective drawings when work is cylindrical, workmen get mixed up on center lines.

Q. 15. What sizes of sheets do you use for drawings?

9"×12" 13
12"×18" 16

18″×24″	20
24″×36″	19

There seems to be little uniformity in the sizes of shop drawings, about 67 firms reporting different combinations. A few have no system but simply make the size of sheet to suit the object to be drawn.

Q. 16. Do you use red ink on tracings?

Never use red ink on tracings	57
Recently discarded the use of red ink	2
Use red ink for pattern figures	1
Use red ink for center and dimension lines	8
Use red ink for check marks	1
Use red ink for existing work on studies	1
Use red ink sometimes	2
Use red ink on occasions when it is desired to show old work in red and new work in black (use carmine)	1
Use carmine for brick	1

Qs. 17 and 27. How indicate finished surfaces on drawings? When finished all over? When "file finished," ground, planed, bored, drilled, etc.?

Finished surfaces indicated as in Fig. 1	65
Finished surfaces indicated as in Fig. 2	16
Finished surfaces indicated as in Fig. 3	8
Finished surfaces indicated as in Fig. 4	2
Finished surfaces indicated as in Fig. 5	2
Bound the surfaces with red lines	2
Bound the surfaces with dotted lines	2
Name the finish by note in full	68
Do not specify machinery method (See drawing.)	6

Q. 18. Do you use horizontal or sloping lines for convention in screw threads?

Sloping lines, see Fig. 6 94
Horizontal lines, see Fig. 7 12

FIG. 1.

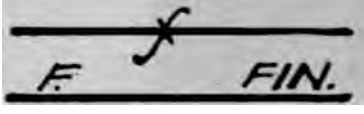


FIG. 2.

FIG. 3.

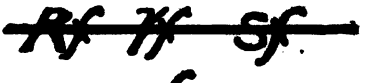
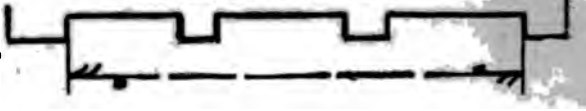


FIG. 4.



Finish only third line from top

FIG. 5.



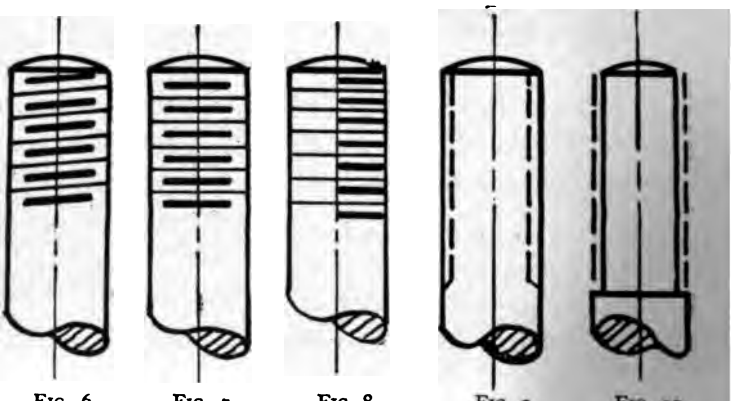


FIG. 6. FIG. 7. FIG. 8. FIG. 9. FIG. 10.

Horizontal lines, see Fig. 8 13
Both 7
Neither, but as shown in Fig. 9 1
Neither, but as shown in Fig. 10

Q. 19. When a large surface is in section do you hatch-line around the edges only?

Hatch-line edges only	62
Sometimes	3
Hatch section all over	54
Do not use hatch lines; shade the whole surface with 4H pencil	3
Usually show a broken surface line	1

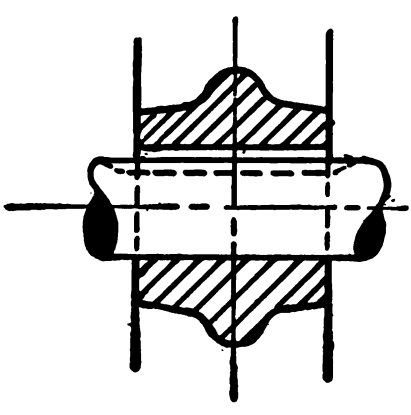


FIG. 11.

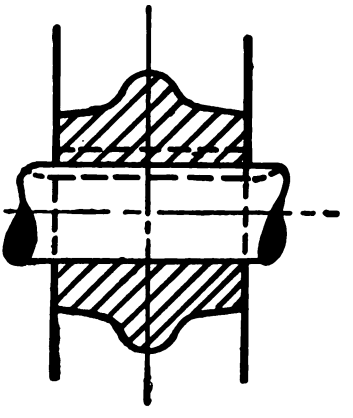


FIG. 12.

Q. 20. Do you section keyways in hubs or show by invisible lines?

Section keyways as shown in Fig. 11	73
Show keyway by invisible lines, see Fig. 12	40
Keyways in hubs left blank	1

Q. 21. In dimensioning do you prefer to place the dimension upon the piece or outside of it?

Outside whenever possible	92
Upon the piece	13

th, according to size and shape of part 19
No rule 1

Commenting on placing dimensions outside of piece one says, "It entails less confusion to workman." Another says: "So as to make detail stand out."

Q. 22. Do you use feet and inches over 24 inches?

Yes 69
Use feet and inches o 4
Use feet and inches ov on foundations and outlines .. 2
Use feet and inches ove 6
All inches 21
For pulleys use inches 1
Inches up to 10 feet 2
Start feet at 24″ thus: _ 2
Usually, but not always..... 2
Yes, except pitch diameters of gears, which are all given in inches 2
Yes, except in boiler and sheet iron work 3
Use feet and inches over 12″ 6
Inches up to 100″ 3
Inches up to 60″ 1

Q. 23. How do you indicate feet and inches? Thus 2 ft. 4″, or thus 2-4″?

2-4″—97, 2FT. 4″—5, 2 FT. 4″—2, 2ft. 4″—13. Both 2ft. 4″ and 2-4″—1, 2FT. 4 IN.—1, 2′ 4″—8, 2=4=—1.

Q. 24. Do you dimension the same part on more than one view?

One view 94
More than one view as check 46

Q. 25. When several parts of a drawing are identical would the dimensioning of one part suffice for all, or would you repeat the dimension on each part?

One part only 82
Would repeat or indicate by note 39
"Left to judgment of draftsman" 1

"When it is evident that several parts are identical the dimensioning of one part would suffice, 'Would never leave room for doubt.'"

Q. 26. Do you write R for radius or RAD.? D. for diameter or DIA.?

RAD .. 35 Rad ... 47 R 32 rad. .. 1 r 3
DIA .. 41 Dia .. 48 D 15 d 3 dia ... 4
DIAM 1 Diam. ... 3 diam 5

Do not use R. or RAD., dimension only 1

Q. 28. Do you always give number of threads per inch? When you do how are they indicated?

Only give number of threads when not standard 67

All others always indicate number of threads in a great variety of ways. A few of the different styles of noting the threads are given below:

$\frac{1}{4}$"—10 Thr. 5THDS. PER 1". 8thds. 4 threads per inch. Mach. Screw 10–24, $1\frac{1}{4}$" XII, 16 P. RH. Vth. U. S. S. XVIII, 1"–8–U. S. S. 1" TAP, 8 PITCH, 3 TH'D R. H. SQ. DOUBLE, 5"–18 THDS. R. H. OWN ST'D 10 thds. per inch. For pipe tap thus, $\frac{1}{2}$" P.T., etc., etc.

Q. 29. How do you "Mark" a piece to indicate on the bill of material?

Number it on drawing and put a circle around it 34

By name or letter 35
By pattern number 2
By symbol and number 14
Castings, I, II, III, Forgings, 1, 2, 3.

Q. 30. When a working drawing is fully dimensioned why should the scale be placed on the drawing?

For convenience of drafting room 25
Check against errors 11
Not necessary 18
Scale not placed on shop drawings 18
For convenience in calculations and planimeter work 1
To give an idea of over-all dimensions when these are not given. "We never saw a drawing so fully dimensioned as to warrant leaving off the scale" 2

"If a drawing is to scale the scale should be on the drawing, whether it is needed or not."

"It gives every one interested a better conception of the proportions of the piece, and there are frequently portions of a design which do not require a dimension for the shop to work to, and which it is interesting to scale from an engineering point of view."

"To get approximate dimensions not given on drawing."

"Impractical to dimension all measurements for all classes of work."

"Scale will tell at a glance, dimensions would have to be scaled."

"To obtain an idea of relative size of parts without scaling the drawings."

"To sketch on clearance." "To proportion changes." "When erecting to measure over-all sizes."

"In case a dimension has been left off, the scale will help out."

"This is a question of opinion; some will not have the scale, others insist on it." "We always give the scale."

"It is an immense help and time saver in the drawing room."

"Generally no reason. In our work we combine standard apparatus by 'fudge' tracing, and it is convenient to know scale so all parts will surely be to same scale."

"In discussing alterations, additions, clearances, etc., it is convenient to know the scale instantly."

"For convenience in drafting room. We often put an arbitrary scale on with a reference letter indicating scale to draftsman."

"To give toolmaker an idea of the size of the finished piece."

"As an aid to the eye in reading."

Above are some of the reasons given for placing the scale on the drawing. Below are given a few of the reasons why some do not place the scale on the drawing.

"Scale should never be used in shop," says one.

"Not necessary. Sometimes drawing is made out of scale."

"Not advisable, on account of workmen getting into the habit of working to scale instead of to the figures."

"Know of no good reason at all."

"Believe it best to leave scale off."

"Should not. Drawing should never be scaled."

"Know of no good reason why it should be."

"Should not be given on drawing."

"Do not object if left off, not needed."

Q. 31. Do you use the glazed or dull side of tracing cloth?

Dull side ... 66 Glazed side. 32 Both 4

"Dull side, because it lies flat better in drawers."

"Dull side, so that changes which may be necessary while work is under construction, can be made easily in pencil and later in ink."

"Dull side so tracings may be checked in pencil."

"It prevents curling."

"Both, although the glazed side when traced on lies better in the drawer."

"We use cloth glazed on both sides, work on convex side, so that shrinkage of ink will eliminate camber."

"Dull, except for U. S. Government, who requires the glazed side to be used."

Q. 32. How do you place pattern numbers on castings?

Pattern number with symbol or letter is placed on or near the piece, e.g., PATT.–D–478–C..... 36

This question was not happily stated; most answers gave "raised letters cast on," while the question like all the others refers to the marking of the drawing.

Q. 33. How do you note changes on a drawing?

On tracing with date..... 32
New tracing and new number..... 17
Put a circle around old figure and write new figure beside it with date..... 8
Make new tracing..... 5
Red ink with date..... 8
Use rubberstamp "Revised" with date, and indicate changes on record print..... 28
Use change card system..... 1
Special forms for purpose. Change made in a book with date. New prints made to replace. In place at title with draftsman's initials and date..... 8

Q. 34. Do you place dimensions to read from bottom and right hand, or all to read from bottom, or how?

Bottom and right hand... 103 From bottom only..... 2
No fixed rule..... 2
From R to L and bottom to top..... 1



Q. 35. Do you always make a table to contain the bill of material?

Yes.....	49	No.....	25	Not always..	5
Usually.....	1	Use separate bill			32

Bills on general drawings only. On details number is marked on piece.

"No, but it is advisable to do so." "Have abandoned that system."

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